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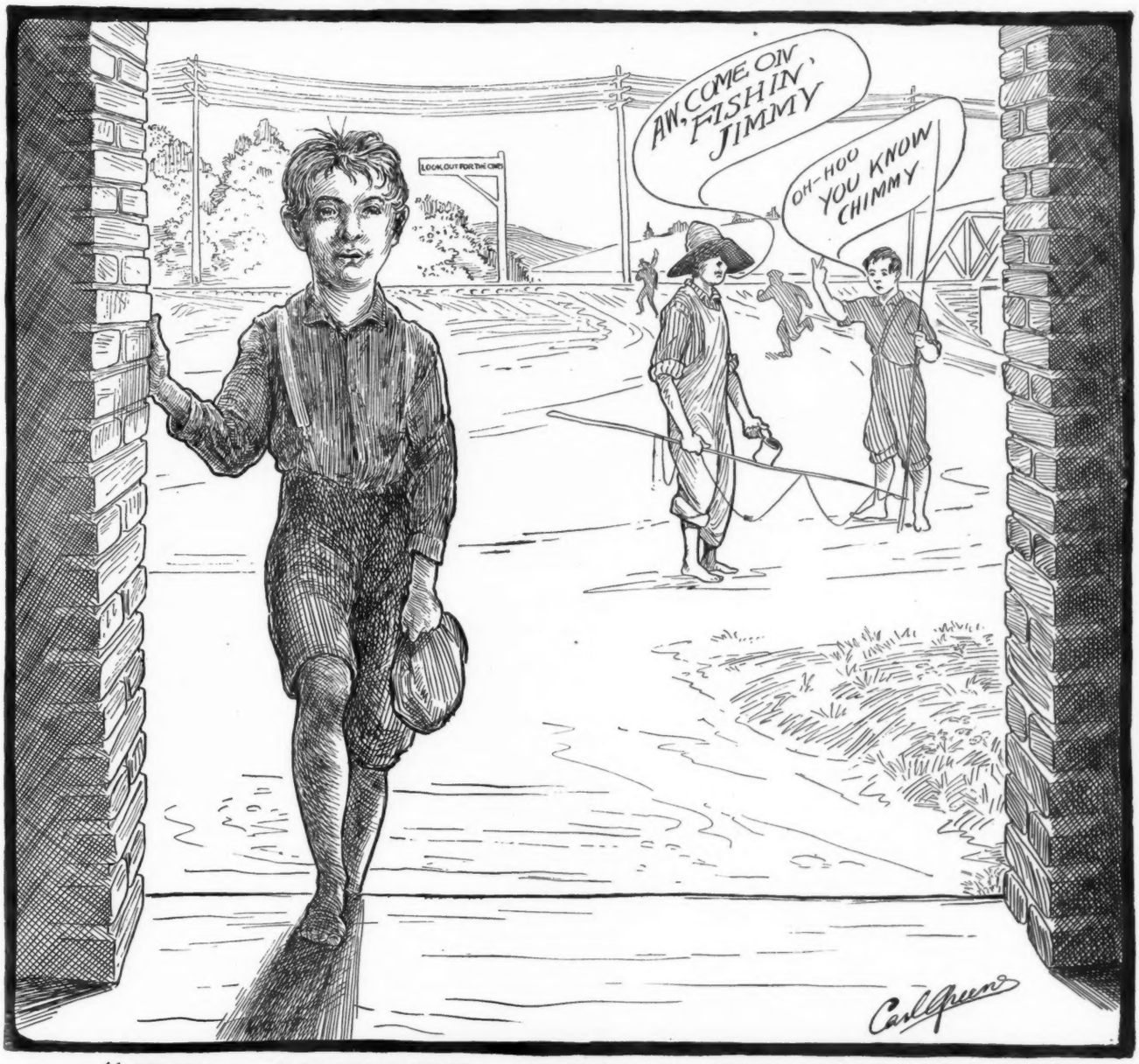
THE BEST RAW MATERIAL



"SAY, MISTER, CAN I COME IN?"

Tests of Wood, Paper and Steel Pulleys

By H. A. Woodworth

Account of tests made by E. D. Biggs and the author at Purdue University in 1909 to determine the breaking strength of pulleys of different kinds with a view to disclosing their principal points of weakness.

The object of the following tests was to determine the relative bursting speeds of different types of pulleys and, if possible, whether the weak points in each form of construction correspond with previous experiments.

The condition of stress existing in the rim of a flywheel was first discussed by J. B. Stanwood (*Trans.*, A. S. M. E., 1893). He showed theoretically that the bending due to centrifugal force might very materially reduce the bursting speed. This was later proved experimentally by Prof. C. H. Benjamin (*Trans.*, A. S. M. E., 1899-1912), who tested model flywheels to destruction by revolving them in a bomb-proof case at varying speeds until they gave way. Benjamin's tests proved that joints between arms consisting of a pair of lugs connected by bolts were a great source of weakness. Further to prove these theories, and test the most improved designs of pulleys a series of tests of 24-in. wheels of cast iron, steel, wood and paper were arranged for with E. D. Biggs and the writer in charge. The tests were conducted at Purdue University in the spring of 1909.

Fig. 1 shows the testing plant which consisted of a cylindrical pit in which was placed a 3-in. vertical shaft driven by a vertical motor. The shaft was supported in the center of the pit by cast-iron bearings bolted to two horizontal channel irons, the ends of which were imbedded in the concrete that formed the walls of the pit.

The pit was 7 ft. deep and 8½ ft. in diameter. The shaft was belt-driven from a 10-hp. variable-speed motor which could be operated from 800 to 2400 r.p.m. Sand was piled around the inside of this pit to prevent flying pieces from coming in contact with the concrete wall when failure occurred. By this means the original fracture was obtained.

The starting box, or controller for varying the speed of the motor, was placed in a safe position some distance from the pit. The tachometer was placed on a heavy wooden block, about 4 ft. from the pit. This block was imbedded in the ground, with its top on a level with the top of the pit.

To determine the speed ratio of the main shaft to that of the tachometer a series of simultaneous readings were taken of the main shaft and tachometer respectively, and from these data a curve was plotted with the tachometer readings as ordinates and the main shaft revolutions per minute as abscissas. This calibration curve was determined for every three or four pulleys tested, and proved to be a straight line through the origin upon which the results of all subsequent tests fell.

After balancing the pulley, photographing, weighing and running the calibration curve, preparations were made to test the first wheel.

This was placed on the shaft in the testing pit, and the pit was covered with planks to prevent any fragments from escaping. The wheel, which will be referred to hereafter as test wheel No. 1, was a two-arm solid-rim built-up type wooden flywheel, shown in Fig. 2 and weighed 29.37 lb. It was strengthened by six ½-in. bolts, four of which were placed at the hub and furnished means of clamping the flywheel to the shaft, the other two being placed one at each end through the continuous arm.

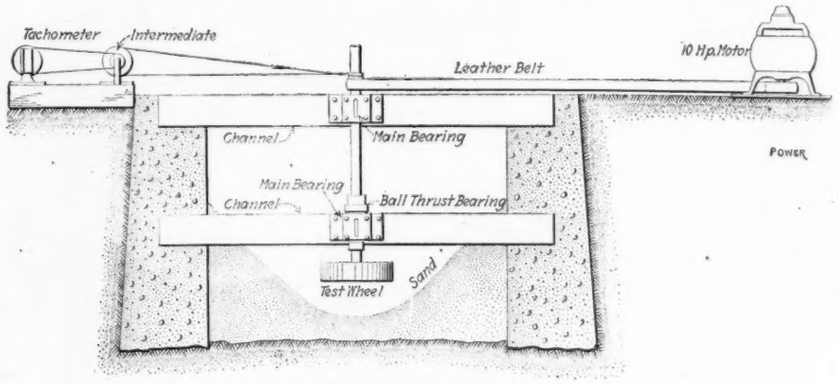


FIG. 1. PIT FOR TESTING PULLEYS TO DESTRUCTION

The test started at a very low speed and as each new speed was obtained the operator stepped the motor to the next speed, using great caution to gradually accelerate the pulley. When the pointer indicated a speed of 2720 r.p.m., there was a noise like the report of a muffled powder explosion as the pulley let go. A fine débris of splinters, bolts and nails, together with large pieces of the rim and flywheel arms was found lying all over the bottom of the pit showing the original disintegration fractures due to the excessive speed.

After the explosion the débris was gathered together and assembled as nearly as could be, and a photograph reproduced in Fig. 2 taken of it. The duration of this test was about 12 min. The peripheral speed at the time of fracture was 285 ft. per sec. The failure came very suddenly. The fracture was probably caused by one bolt in each end of the continuous arm giving out due to the centrifugal force. Tabulated data and results of the tests are shown in the accompanying table.

No. 2 was a duplicate of No. 1 and burst at a speed of 2550 r.p.m. or a peripheral speed of 267 ft. per sec. The whole rim was shattered, but the arms remained intact.

No. 3 was a split-rim built-up wooden pulley, the split being made at the ends of the arms, as shown in Fig. 3. This wheel was a duplicate of the first two wheels tested, except that the former two had solid rims. It burst at a speed of 2210 r.p.m. or a peripheral speed of 231.8 ft. per sec. The whole rim was shattered but the arms remained as shown in Fig. 3. A close inspection reveals the uniformity in the strength of the split-rim wheel where the joint is at the end of the arms.

Wheels Nos. 4 to 10 inclusive were duplicates of No. 3. In all these wheels the fracture was due to weakened rims caused by the holes made in them at the factory, in which were placed balancing weights. These weights together with the weakened section caused the failure. In some cases the bolts holding the arms to the two sections of the rim were considerably bent. In general the split-rim flywheel of this type has only 80 per cent. the strength of the same type with a solid rim. The results of the tests are shown in the table.

No. 11 was a solid castiron wheel, shown in Fig. 4. It had six arms, each having a cross-sectional area of 0.99 sq.in. It had a 6-in. face and weighed 70.44 lb. This wheel was held on the shaft by two case-hardened setscrews. Two attempts were made to break it, but each time the fuses burnt out at the switch. The maximum speed attained was 3720 r.p.m., or a peripheral speed of 389.4 ft. per sec.

No. 12 was a duplicate of No. 11. To avoid air resistance, sheet-iron disks were placed on each side of the wheel, and wired together through the spokes or arms. This did not avail anything and the maximum speed obtained was 3380 r.p.m., or a peripheral speed of 353.8 ft. per sec. Further tests of these wheels were abandoned, owing to want of more power, as 15 hp. proved insufficient.

No. 13 was a paper wheel having a solid rim and web, shown before and after being disintegrated in Fig. 5. This wheel was balanced in the web. The rim was held to the web by ⅝-in. wooden pins parallel to the axis of the shaft and approximately 4 in. from center to center. The hub consisted of two heavy cast-iron plates bolted to opposite sides of the center of the web forming the hub.

This wheel burst at a speed of 2820 r.p.m., a peripheral speed of 295.2 ft. per sec. The fracture occurred as anticipated. A part of the rim 1 ft. long flew off, separating itself into one large piece and several small fragments. The wooden pins connecting the rim to the web were broken off squarely. It is thought the fracture was due to the heaviness of the rim and the weakness of the pins connecting it to the web.

No. 14 was a duplicate of No. 13, and broke in the same manner. The fracture was presumably due to the same cause. These two tests demonstrate the lack of strength in the rim of this type of wheel, which suggests a field for future improvement.

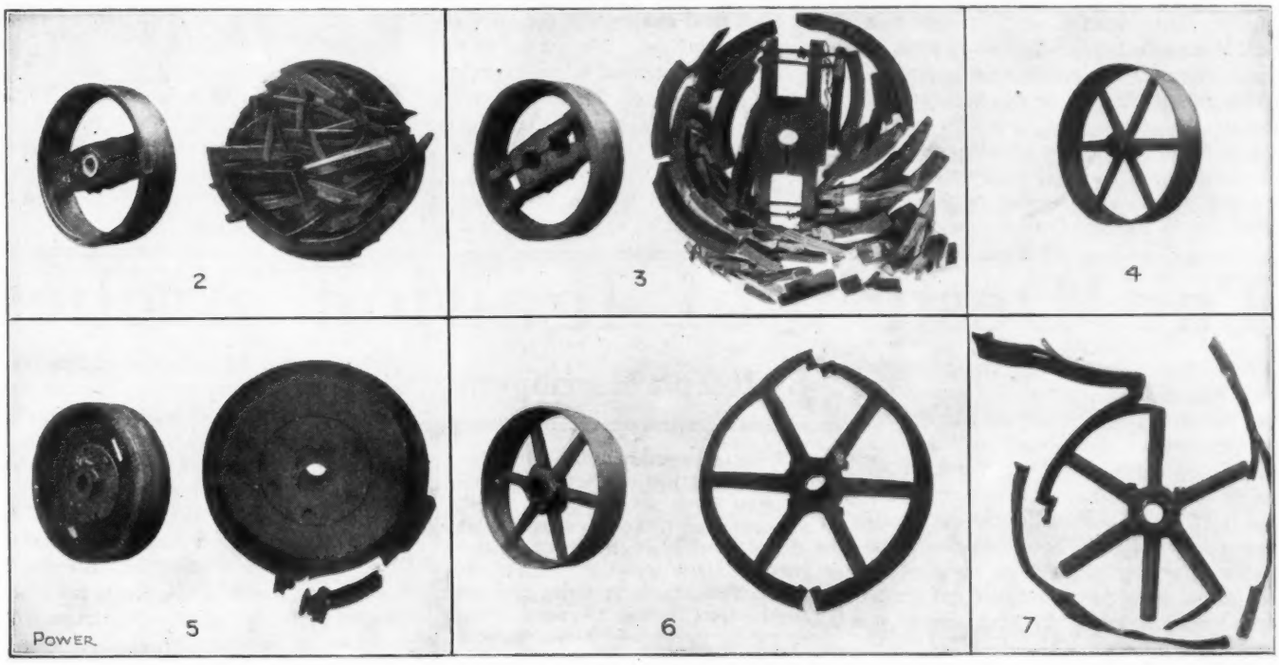
No. 15 was a pressed-steel split wheel; that is, it consisted of two sections or halves.

Fig. 6 shows this wheel before and after fracture had occurred. In this type the rim was riveted to the arms, of which there were six. The sections of the rim were fastened together by a small flange which was riveted at each joint to the rim. A ¼-in. bolt connecting these flanges held the two sections of the rim together. Each arm consisted of two flat pieces of iron 1¾ x ⅜ in. in cross-section, riveted one on each side of a seam on the inside of the rim. This type, having a very light rim, should have stood a very high speed, but it was seen that the rim was very weak at the joints.

The ends of each half of the rim where they were joined together were bent outward, thus rupturing the small flanges and bending the bolts that connected the two small flanges. The instant that a rupture was suspected the power was shut off, and when the motor stopped the rim was found bent out of shape, but not torn from the arms. The speed obtained was 2240 r.p.m., or a peripheral speed of 234.5 ft. per sec.

No. 16 was a duplicate of No. 15. The first fracture occurred in the same manner as with No. 15. Instead of shutting down the motor when the first distortion was suspected, it was continued until the greater part of the rim was torn away from the arms which resulted in a loud crash. The rim was torn entirely away from four of the arms, but a part still remained attached to the other two arms. The metal was sheared out, where the flanges were riveted to the rim, but the rim was sheared out where the arms were riveted to it. The speed was the same as for No. 15, being 2240 r.p.m., or a peripheral speed of 234.5 ft. per sec. In this case the bolt failed, as shown in Fig. 7. This illustration is interesting, as it shows clearly the bending action of the centrifugal force on the bolts, using the inner edge of the flange as a fulcrum.

The test of No. 15 was stopped at the critical moment. To have continued it would have resulted as shown in Fig. 6. This indicates how the joints failed. The joint opened like a clam shell, the bolts and flanges sheared out from the rim, and the arms remained uninjured. The joints in Fig. 6 display the shearing action, or cantilever effect, which in experiment 16 indicates that, as the bolts were strained and the joint loosened, one side of the joint was more affected by the centrifugal force than the other, causing a shearing action in the joint. These wheels burst at a much lower average speed than the paper, solid-wood or even split-rim wooden wheels. The test proves the source of weakness in the rim joint midway between the arms.



FIGS. 2 TO 7. VARIOUS PULLEYS TESTED TO DESTRUCTION

The stresses given in column 7 of the table were computed by the formula

$$\textit{Stress per sq.in. of section, lb.} = \frac{V^2}{C}$$

where C is 100 for wood and paper and 10 for iron. The velocity V is the linear velocity of the center of gravity of the section. The values worked out in the table show to what extent the strength is affected by bending. The above formula has been shown by Mr. Stanwood to represent approximately the tensile strength per square inch of rim section due to centrifugal force.

In every case the rim demonstrates the faulty design followed in modern practice. The surprisingly low bursting speed of the steel wheels proved conclusively

that rim joints midway between the arms should be avoided and these joints placed at the ends of the arms. The strength of the joint at the end of the arm is shown in the tests of the solid-rim wooden wheels, Nos. 1 and 2 as compared with the split-rim wheels Nos. 3 to 10 inclusive. These were duplicates, except that one had a split rim and the other a solid rim.

These tests are the first ever made of paper, wood or pressed-steel wheels. In fact, this is the first time the original fracture of a wheel tested to disintegration has been determined. In all previous tests the casing was so designed that flying particles struck it with such force, that they were again broken, making it impossible to reassemble the pieces. It was due to the remote control that we were able to obtain the bursting speed so accurately.

Balancing wheels by adding weight in the rim between the arms produces a cantilever effect. In the iron wheels a piece of iron the size of a goose egg is sometimes riveted to the rim between the arms to balance it. At very high speeds the bending, due to the centrifugal force of this weight, causes disintegration. The wooden wheels are balanced by inserting iron lugs in the wooden cross-section of the rim which decreases the cross-sectional effective area, weakening the rim, and failure occurs at this point. Wheels should be balanced by weights at the ends of the arms.

Conclusions

(1) Balancing the wheels in the rim causes fracture at low speeds and thereby lowers the factor of safety.

(2) Rim joints midway between arms are serious defects and materially reduce the bursting speed.

(3) The solid-web-and-rim paper flywheel of the same type as tested (Fig. 7-8) will safely withstand a rim speed of 106 ft. per sec. By properly strengthening the rim the speed may be materially increased and a sufficient factor of safety retained.

(4) Wood flywheels with solid rim (Fig. 2) have an ample factor of safety at a rim speed of 90 ft. per sec. if the wood is of good quality and the same design is followed out. The speed of this wheel may be increased by using lighter bolts at the ends of the arms, as the weight of the bolts caused the original fracture in 90 per cent. of the wheels tested.

(5) Wood split wheels of the same design and material as shown in Fig. 3 will have a sufficient factor of safety at 72 ft. per sec. rim speed. The design may be improved by using lighter bolts and discontinuing the practice of balancing in the rim.

(6) The iron flywheels, Fig. 4, which were not tested to destruction, if of good iron and free from serious cooling strains will be safe at a rim speed of 120 ft. per sec. These wheels stood the best test of any of which there is record. The bending in the rim, due to centrifugal force, is so slight that it may be neglected in this type of wheel.

(7) Steel wheels of the split-rim type, Fig. 6, are unsafe at speeds above 80 ft. per sec. It has been noted that in various cases wheels of this type gave away through a solid rim and without failure of the bolts, although the strength of the bolts was less than one-third that of the rim section, which shows that the strength of this joint as usually calculated has very little to do with the operative strength of the flywheel.

It is not necessary to go farther into the design and the safe speeds of various types of flywheels, as this subject has been splendidly treated by F. A. Halsey in the Mar. 28 issue of the *American Machinist* in an article entitled "Design of Small Flywheels."

The Design of Ignition Arches

By Joseph Harrington

Principles underlying the design of ignition arches, with reference both to the arch that absorbs heat from combustion of the volatile at the front of the furnace and to that which receives radiant heat from the incandescent fuel at the rear. Flat arches compared with sprung arches.

An ignition arch is a refractory tile roof placed over part of a furnace for the purpose of producing or stimulating the ignition of fresh fuel and is used largely in connection with stokers in boiler furnaces. The form and extent of this arch have undergone radical modifications in recent years and the principles governing its influence on the fuel bed were formerly considered quite different from those which now appear to control its effects.

A few years ago ignition arches were built as short as 2 ft. and were seldom over 4 ft. in length. They were placed at slight inclines to the horizontal with the rear tilted upward, and their height was measured from the grate surface without reference to the distance from the surface of the fuel to be ignited. The actual distance was slight, the under surface of the arch being placed but 2 or 3 in. above the fuel.

The volatiles, liberated by the heat of the furnace, ignited at the surface of the coal and burned in contact with the under side of the ignition arch. They imparted some of their heat to the firebrick which was raised to a state of incandescence and radiated heat to the incoming fuel with sufficient rapidity to perpetuate the ignition and render the operation continuous.

No account whatever was taken of the temperature produced by the combustion of the fixed carbon at the rear of the furnace, and that is just where the most desirable source of heat was neglected. One thing that developed in the writer's experience with radiating arches of uniformly heated surface was their observance of the following laws:

1. The maximum intensity of radiation was perpendicular to and at the center of the radiating surface.

2. The temperature of any point within the affected zone was inversely as a power of the distance from the incandescent surface, disregarding the influence of the fuel bed or burning volatile.

A recognition of these two facts helped in many places where ignition was poor or uneven. The first led to a horizontal arch set to throw the center of radiation farther forward, and thus increase the effectiveness of the radiation, and the second is the reason for the greater uniformity of igniting effect with flat arches as compared to sprung arches. Fig. 1 shows the relative effect of a variation in the inclination of the arch and Fig. 2 the variation in the temperature under a sprung ignition arch. The sketch A, Fig. 1, illustrates the heat distribution with a horizontal arch, the semicircular shaded portion showing approximately the variation in intensity of radiation. It will be noted from this that the area subjected to radiation includes practically the entire extent of fuel surface directly beneath the arch. Sketch B illustrates the variation in the extent and intensity of the radiation when the arch is tilted, and sketch C the still greater reduction in ignition effect when the arch is tilted still more.

Fig. 2 is a sectional view with lines to indicate the variation in temperature at different points in the furnace, the lines *a*, *b*, *c* and *d*, defining zones of equal temperature and consequently of equal ignition effects, due to the heat radiated from the under surface of the arch and the adjacent side walls. It will be observed from this that those portions of the fuel bed nearest the sides are subjected to a greater intensity of ignition than is the center of the fuel bed. The result of this is the more rapid ignition and consequently earlier completion of combus-

tion along the sides than at the center. The rapidity of ignition is, of course, in direct proportion to the temperature, so that with such an arch, the combustion proceeds more rapidly at the sides than in the middle. This undesirable effect is offset somewhat by making the arch as flat as possible, but the influence of the curvature remains, and on wide furnaces is serious. A flat arch provides a uniformity of radiation and the ignition effect is of necessity the same for the entire width of the furnace.

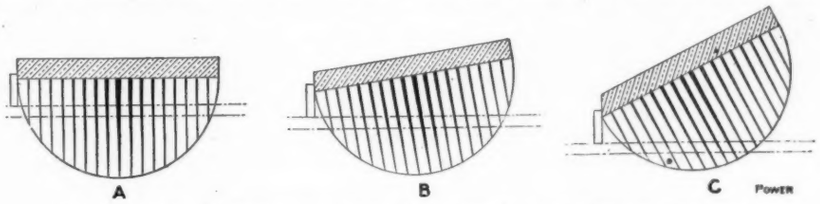


FIG. 1. HEAT DISTRIBUTION WITH HORIZONTAL AND INCLINED ARCHES

In accordance with the second law of radiant arches, the nearer the incandescent surface of the arch approaches the

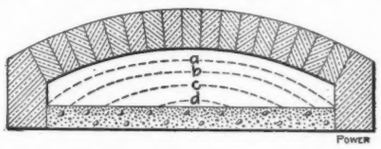


FIG. 2. VARIATION IN TEMPERATURE AT DIFFERENT PARTS OF FURNACE

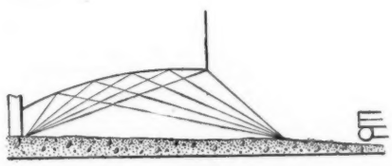


FIG. 3. REFLECTING ARCH FOR TRANSFERING HEAT FROM BACK TO FRONT OF FURNACE

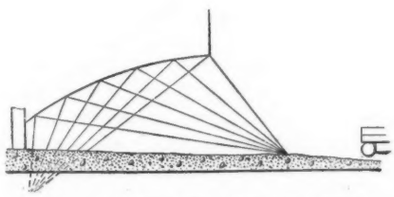


FIG. 4. ARCH SET OUT OF FOCUS

fuel to be ignited, the greater will be the intensity of the ignition. It is obvious that when the distance is but a few inches, that a slight variation will decidedly affect the influence of the arch. It has heretofore been the common practice to build these arches at a definite distance from the grate surface, no account being taken of the location of the surface of the fuel bed. It is quite obvious, however, that this is or should be the deciding factor and the arch can only be intelligently set when the thickness of the fuel bed is predetermined.

The writer early gave this matter consideration and has followed the practice of setting igniting arches purely with reference to the surface of the fuel bed and not with reference to the grate surface. On the other hand, there is a certain limitation to the proximity to the fuel bed with which an arch can be set. There must be sufficient space to permit of combustion of the volatile matter and to permit the free flow of gases away from the fuel bed. Otherwise, the combustion is smothered and the value of the arch is entirely lost.

All of the foregoing discussion has related exclusively to that type of arch designated as radiant or those arches which after absorbing heat from combustion of the volatile constituents of the coal have become incandescent and are then in a condition to radiate heat to the incoming fuel.

As suggested, no attention had heretofore been paid to the influence of the rear end of the fuel bed. The development of new types of stoking apparatus and the necessity of igniting various grades of coals resulted in experimentation with arch slopes and lengths in connection with various kinds of fuel and various types of stokers. During the course of this work, it became apparent that there was another source of heat than the combustion of the volatile at the front of the furnace and that this source of heat was far more reliable and abundant than the other. Attempts were then made definitely to utilize this heat by means of the ignition arch, which was set in various positions and the degree of igniting effect carefully noted. It was early apparent that the old ideas concerning these arches were only half correct and a new theory as well as a new practice has been the outgrowth of such experience.

An analysis of the setting of an arch successfully utilizing the heat from the rear of the furnace showed that it was set just as if a ray of light was to be reflected from the rear to the front; see Fig. 3. The rays of heat follow exactly the well known law of the reflection of light, namely, that the angle of incidence equals the angle of reflection.

Space will not permit setting forth all the considerations which govern the correct design of a reflection arch. The essentials can be worked out mathematically and with a given length of fuel bed and thickness of fire, an arch can be designed that will exactly reflect heat from the rear or incandescent portion of the fuel bed to the point of entrance of the green fuel. Knowledge of the location of the center of supply with various grades of fuel, the effect of varying percentages of moisture and ash on the temperature of the glowing coke and the rate of absorption of heat by the fresh fuel must be had. There are many combinations of these variables which, taken with the influence of the surrounding brickwork, make the exact locating of a reflecting arch a complex problem. Therefore, the following treatment of this subject will be confined to the simple problem of a clean, free-burning coal on a chain grate.

Fig. 3 shows a reflecting arch which is designed to transfer the heat from a single point in the incandescent fire to a single point in the front of the furnace, and is made to illustrate the principle upon which the entire subject is based. The arch surface assumes a concave form and is indeed nothing more than a condensing mirror with the focus close to the junction of the fuel surface and the stoker gate. Such an arch is undesirable, not only from a construction standpoint but because it concentrates the reflected heat too much.

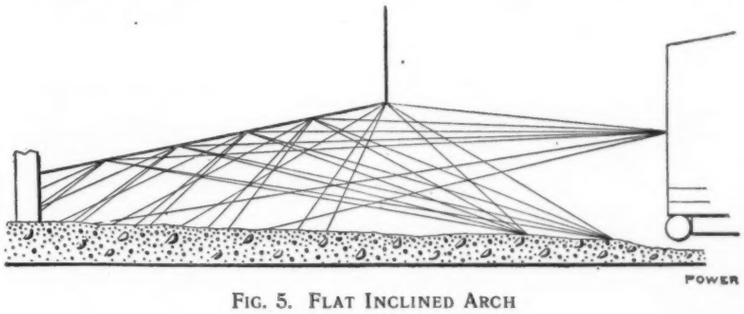


FIG. 5. FLAT INCLINED ARCH

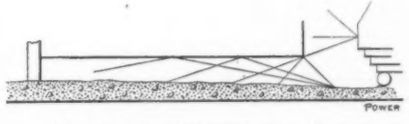


FIG. 6. LOW HORIZONTAL ARCH

The igniting influence must be of some duration in order to thoroughly inflame a reluctant fuel so that this arch would

be actually of greater effect if it were so set as to be "out of focus" as illustrated by Fig. 4.

The objections to the curved arch can be overcome without any departure from the governing principles by the use of a flat inclined surface; see Fig. 5. In this the heat rays emanating from several points of the incandescent fuel are shown reflected from the arch, striking the fresh fuel near the point of entrance, yet sufficiently distributed to give the required duration.

The ignition arch defines the extent of the zone of ignition of the furnace of which it is a part and just as long as the fuel remains under the arch, it must be considered as still subjected to the igniting effect. An ignition arch may, therefore, be too long. Every inch of arch beyond the amount required for complete and proper ignition is a detriment to the furnace. This, then, becomes one of the matters to which all the conditions of the fuel and furnace contribute and it is one which can only be successfully solved in the light of a varied experience.

The law governing the radiation of heat from an incandescent arch applies equally to an incandescent fuel bed so that the height of the arch above the fuel exerts a material influence on its ability to collect the heat rays from a point at the rear. This is well shown in Fig. 6, wherein the low horizontal arch is well designed to prevent most if not all of the heat at the rear of the furnace from coming forward either directly or by reflection.

Far removed as it may appear, the brickwork of the bridge-wall should exert a considerable influence on the ignition, but the low arch effectually cuts off this desirable heat. This type of arch, therefore, is in no sense reflecting and in spite of its continued use is restricted in its successful application to a few limited conditions of coal and draft. It is herein shown as illustrating a design that has persisted throughout the period of improvement in furnace design. Today, every element contributing to the progressive combustion of coal has been carefully studied. The requirements of each stage are comparatively well known and the heat distribution can be scientifically controlled. The most potent element of such control is unquestionably the reflecting arch.

Safe Piping for Boiler Plants

By David Moffatt Myers

A typical layout in which safety and economy go hand in hand. A feature is two valves between the boiler and header and two valves on the blowoff.

Provision for safety in their boiler plants is shamefully if not criminally neglected by many owners. Every article that is published which is written with a view to improve conditions for safety of workmen and operatives deserves to be read and to be studied in proportion to its authority. The design of steam and blowoff piping in boiler plants, especially factory boiler plants, offers a large field for improvement in reducing danger to the men and at the same time simplifying and bettering the plant from an engineering standpoint. Reference is made to the article bearing on this subject on page 488 of the Oct. 1 issue of POWER.

In the same connection is submitted herewith a blueprint of the steam-piping layout for a plant recently designed by the writer, which consists of four 6½x 20-ft. horizontal tubular boilers to operate at 125 lb. working pressure. In order to increase the safety of the piping to operatives, and to provide for proper drainage and expansion, this plan is submitted for the attention of owners who will not, or think they cannot, afford the services of a competent engineer for designing their piping system. The several points that demand special attention are as follows:

TWO VALVES BETWEEN BOILER AND HEADER

Each lead has two valves between the boiler and the header, a provision that should always be made. The expense of the extra valve is slight as compared to the value of a man's life. If the usual single valve gave way during the internal cleaning or inspection of a boiler, he would be trapped therein and killed by the inflow of steam from the boilers connected to the header. With two valves the man's chances are doubled.

Some insurance inspectors insist on opening a flange union between the stop valve and the boiler, and slipping in a blind flange for their additional protection when in the boiler. Most of them, however, are careless. No doubt familiarity breeds contempt, but it does not lessen the risk.

In the plan given, the valve closest to the boiler is of the nonreturn type and acts much like an ordinary check valve. In case there should be a failure in any boiler in the battery such as the blowing out of a tube, the steam from the other boilers on the line would be prevented from entering the injured boiler. This precaution greatly reduces the extent of possible damage. Furthermore, this nonreturn valve would automatically prevent some careless operative from turning steam into an empty boiler which is off for cleaning or repairs and perhaps has a man working inside.

The construction of the leads from the boiler to the header admit of perfect expansion in every direction, thus avoiding bad strains in the piping. Owing to the location of the steam valves in the horizontal run of the lead, it is impossible for dangerous water to collect in the piping when a boiler is shut down. Any condensation flows either back to the boiler or to the main header.

In connection with the nonreturn valve, the question may be asked why a combination automatic nonreturn and stop valve is not used. The answer is that in seventy-five cases out of a hundred that were recorded by one of the large boiler-insurance companies, only twenty-five out of the hundred operated when they should have done so. In other words, actual experience with these double-function valves has indicated that in general they are not sufficiently reliable to be counted upon in all emergencies. If not used frequently, and carefully watched, they are liable to stick and refuse to work when most needed. Consequently the valve with the simpler mechanism, although of the single function (that of preventing steam from flowing into the boiler) has been selected.

MAIN STOP VALVE

Means for preventing the sudden and dangerous emptying of the boilers by the bursting of an engine steam pipe or the knocking out of a cylinder head are provided at the main stop valve *M*, located between the battery of boilers and the steam main leading to engines and heating lines. This main stop valve is provided with a chain wheel, the chain of which passes around another wheel within easy reach of the firing floor.

In case of a blowout, such as mentioned, the engine-room crew will be free to escape and save themselves. For by spinning around the crank on the lower chain wheel on the safe side of the engine-room wall, the main stop valve can be quickly closed and with safety to all concerned. Fig. 2 shows the means for operating this main stop valve. A reduction gear is not used since quick closing is desired.

EXPANSION AND DRAINAGE

As for the safety of the leads and headers, these are all of extra-heavy

piping. The valves, flanges, tees and other fittings are extra heavy. All elbows have long radius to reduce friction losses and improve expansion. The leads may be equally well run at right angles to the boiler header instead of at an acute angle which was done in this case on account of limited space. The 10-in. boiler header is sloped downward in the direction of the natural flow of the steam so that any condensation will drain with the current. A drip pipe will be taken out of the bottom of the wall header or main which is the lowest part of the boiler-room piping, and the condensation will be caught and discharged by a trap.

The wall header or main which lies at a right angle to the boiler header is anchored to the boiler-room wall. The piping to the left is free to expand without strain, owing to the flexible construction shown, the piping to the right has sufficient expansion by virtue of bends to the engines and other lines. Steam should always enter a header or main from the top and be taken out from the top to engines and mill lines, the headers or mains to be drained as described.

Two Valves on Blowoff

In addition to the above precautions each boiler should have two valves on its bottom blowoff pipe between the boiler and the blowoff header. If there is a surface blowoff pipe connected to the blowoff header this pipe also should have two valves separating it from the header.

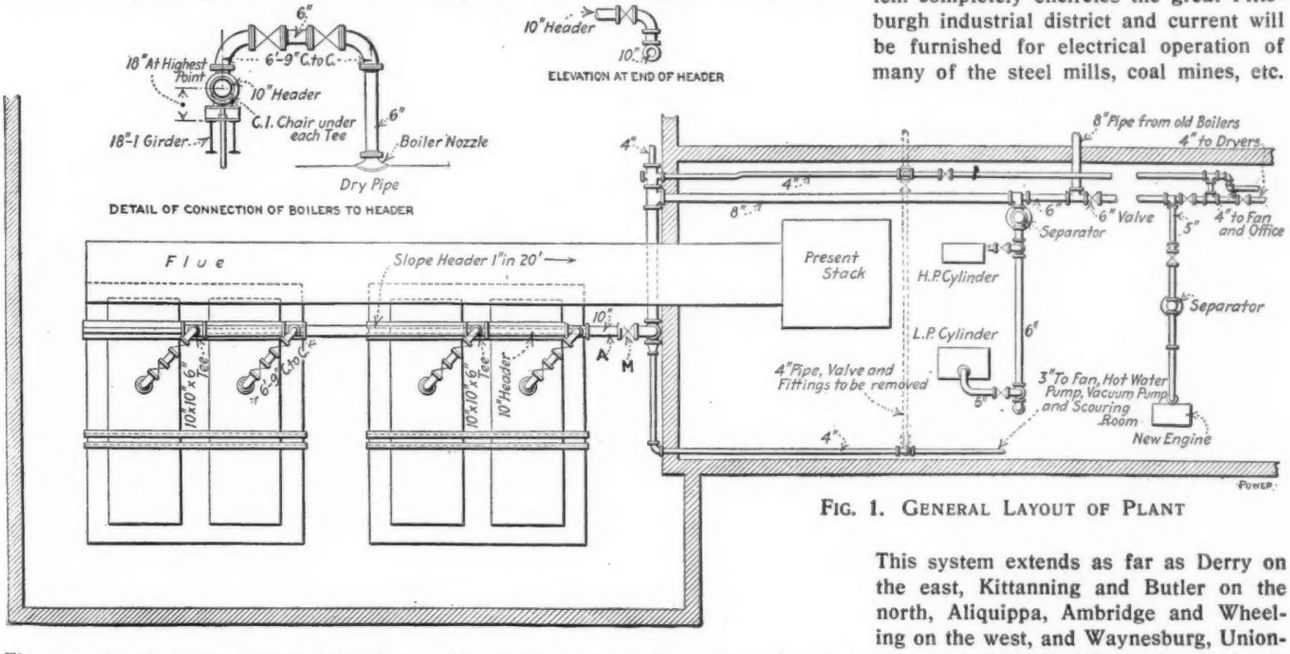


FIG. 1. GENERAL LAYOUT OF PLANT

These extra valves are recommended first for the protection of a man who may be inside of a dead boiler to prevent his injury or death in case of the possible giving way of a single valve for either of these purposes. Secondly, these extra valves are economical from an engineering standpoint as they halve the possibility of trouble and loss by leakage which is most common to blowoff valves.

Safety and Efficiency

There is one thing more that requires emphasis. When a man is inside of a boiler, it should be made an impossibility for any thoughtless or vicious

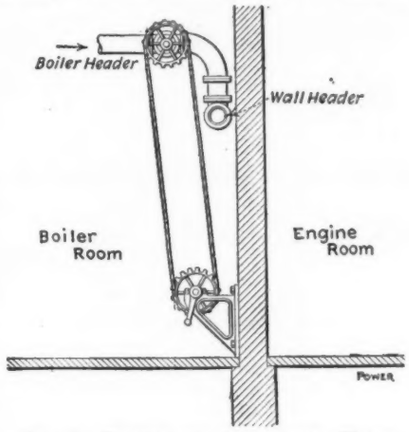


FIG. 2. DEVICE FOR OPERATING MAIN STOP VALVE

person to turn hot water or steam into that boiler. This can be accomplished by padlocking the steam and blowoff valves, and then by giving the keys of the padlocks into the keeping of the man who is to enter the boiler.

The insurance inspector has required the insertion of a connection for the fire pump at *A* in Fig. 1, so there would always be available a supply of steam for this purpose even when the main stop valve might be closed.

Perhaps it will be noticed by some readers how strangely coincidental seem to be the requirements for safety with those of good engineering in steam-piping practice. Stepping up to a broader view, which includes a study of the whole plant, this same coincidence of engineering economy and welfare of operatives is found by the investigator. The more comprehensive the study, the more strikingly will it be found that the highest economic efficiency obtains in those plants or industries where brotherly consideration of the welfare of the workers is the ruling spirit in the mind of owner and manager.

It has long been in the minds of many that there may be an essential underlying and close connection between scientific and moral laws. If moral laws do not sufficiently appeal to some owners and managers to protect their boiler piping and workmen from accident, they will be wise to heed the laws of science and engineering in order to insure their personal profit and to improve the efficiency of their business.

New Cheat River Hydroelectric Plant

The West Penn Traction & Water Power Co., Pittsburgh, has completed arrangements for the equipment of its new 40,000-kw. hydro-electric plant, the first of a series of developments on the Cheat River, at the Pennsylvania-West Virginia line. The electrical apparatus includes four 10,000-kw. Allis-Chalmers vertical waterwheel type alternators, two 300-kw. motor-generator exciter sets and accessories of the same company's design.

The territory to be served by this system completely encircles the great Pittsburgh industrial district and current will be furnished for electrical operation of many of the steel mills, coal mines, etc. This system extends as far as Derry on the east, Kittanning and Butler on the north, Aliquippa, Ambridge and Wheeling on the west, and Waynesburg, Uniontown and Connellsville on the south. It involves 259 miles of 110,000-volt lines.

The Port Arthur Light & Power Co., Port Arthur, Tex., recently organized with a capital stock of $600,000, has taken over the property of the Port Arthur Traction Co., says *The Tradesman.*

Small Westinghouse Steam Turbine

The steam turbine has won its greatest success in large capacities but its phenomenal advancement and the concurrent development of rotary pumping and blowing apparatus, and of electric generators of high rotative speed, has extended and emphasized the demand for units of smaller capacity.

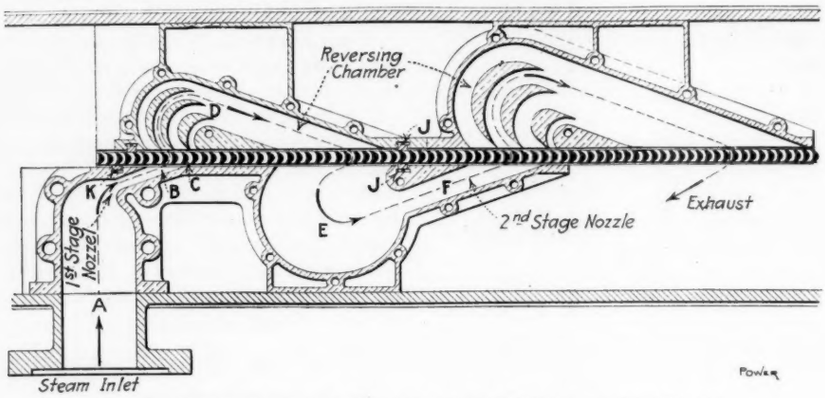


FIG. 1. DIAGRAM OF THE COURSE OF THE STEAM IN THE SMALL WESTINGHOUSE TURBINE

should run at speeds allowing them to be directly connected to such apparatus, and which without multiplication of runners, should be able to abstract a large proportion of the energy due to the considerable pressure- or temperature-fall available.

Since the total amount of steam required by a turbine of small capacity is not greater than can be discharged by a few nozzles of moderate size occupying only a small portion of the periphery of the runner, and since the steam must be made to act a number of times upon a wheel revolving at any practicable rim velocity, in order that the energy induced by expansion between the limits usually available may be absorbed, the natural impulse is to make the steam act again upon that portion of the wheel not occupied by the initial nozzles. This method is described in the patent of Wilson (1849) and that taken out by Ferranti prior to 1870, and successfully used in the Elektra turbine (see POWER, Apr. 6, 1909). It is shown diagrammatically as used in the small Westinghouse turbine in Fig. 1.

Steam enters the chamber *A*, is expanded in the nozzle *B*, passes through the blades *C* of the revolving wheel, is reversed in the chamber *D*, redirected upon the wheel, as shown by the dotted arrow line, and when it arrives at the chamber *E* has been brought nearly to rest. It is then expanded again in the second-stage nozzle *F* and its velocity abstracted by two more passages through the wheel as before, coming out finally into the casing at *H*, with its velocity again largely abstracted, and ready to

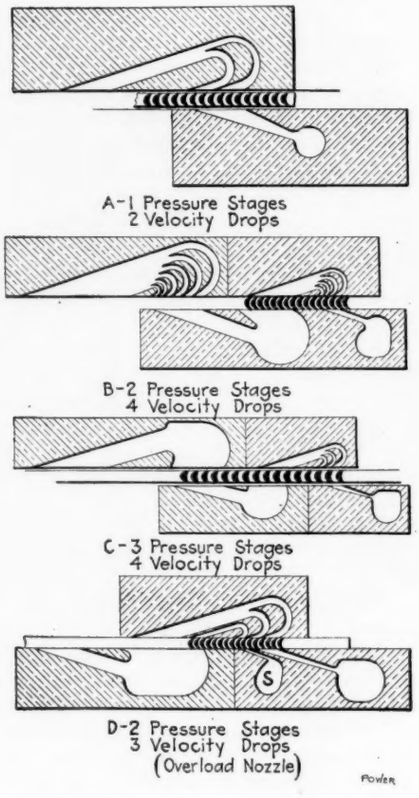


FIG. 2. VARIOUS COMBINATIONS OF PRESSURE AND VELOCITY STAGES

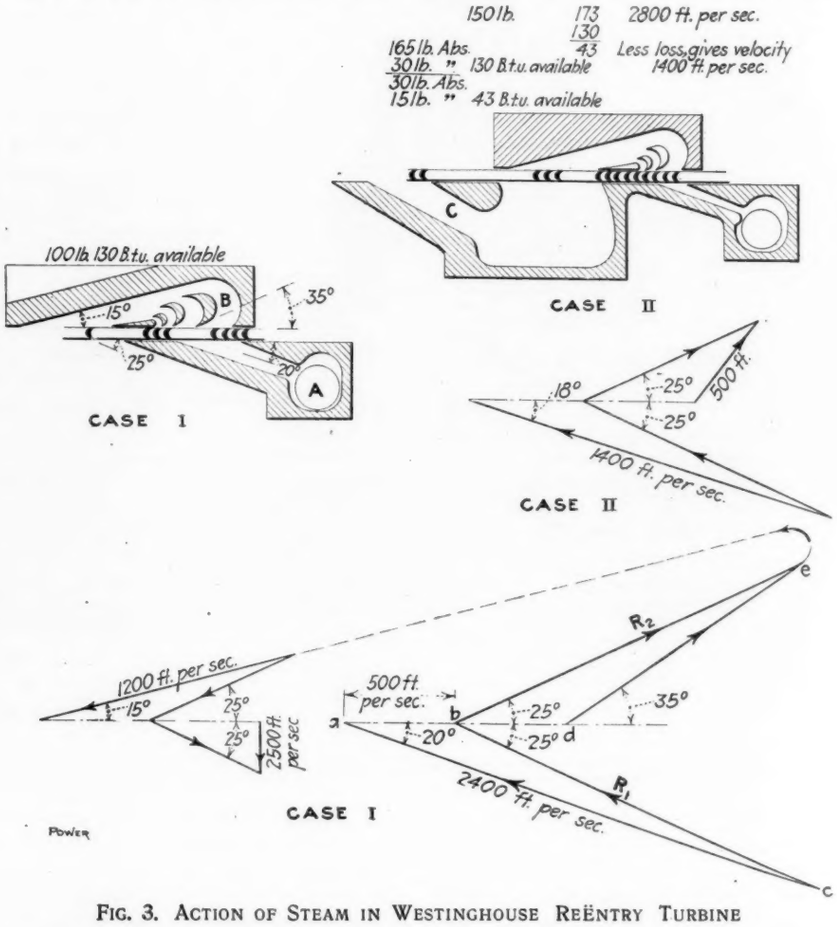


FIG. 3. ACTION OF STEAM IN WESTINGHOUSE REËNTRY TURBINE

Numerous makers have attacked in various ways the problem of producing small, compact and simple turbines which

find its way to the exhaust outlet. Guide frames are used in the reversing chambers to redirect the steam without shock or eddy. A packing is provided at *J* to prevent leakage between stages and at *K* to prevent leakage from the high-pressure stage into the casing. The wheel runs in steam of the exhaust pressure and low density with less windage than as though it were in contact with the denser steam of the initial pressure, as is the case with the disk in the initial stage of a multicellular machine.

It is apparent that this arrangement permits the designer considerable latitude in the choice of stages both pressure and velocity and that to a given rotor in a given cylinder or casing any number and variety of nozzles may be fitted, up to the limit of the blade circle. For units under 50 hp. the descent from boiler to exhaust pressure is made in one expansion stage and the velocity thus generated is taken out by two passes through the wheel. Such a turbine would consist of a single nozzle and reversing chamber, as shown in the upper sketch in Fig. 2, the small amount of steam involved usually not warranting the complication even of the second-stage nozzle and reversing chamber.

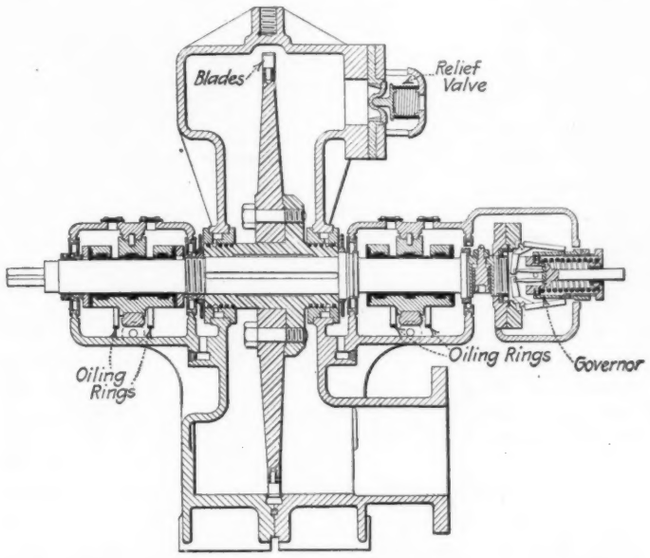


FIG. 4. THE POWER TYPE

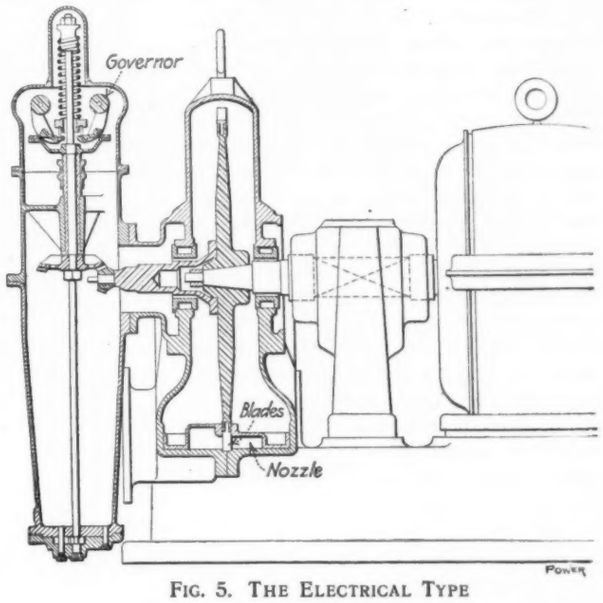


FIG. 5. THE ELECTRICAL TYPE

The nature of the problem will be apparent from the consideration of a couple of concrete cases.

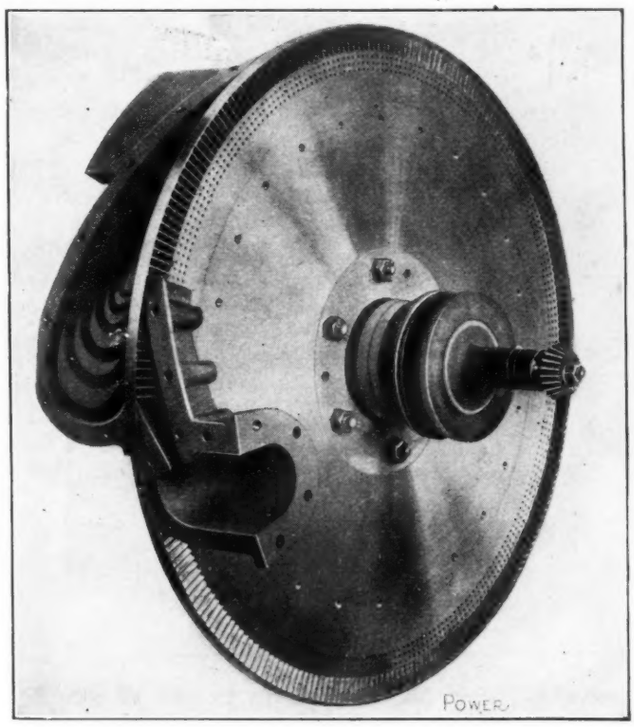


FIG. 6. WHEEL, NOZZLE AND REVERSING CHAMBER OF A NON-CONDENSING TURBINE

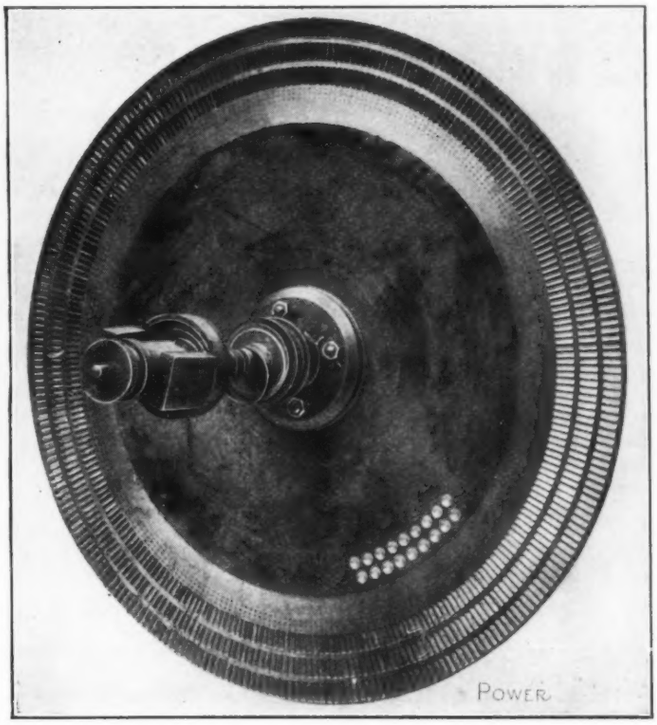


FIG. 7. THE RUNNER OF A 450-HP. CONDENSING TURBINE, 1000 R.P.M.

CASE I: Assume 100 lb. initial and 15 lb. final pressure both absolute, steam initially dry saturated.

This drop makes available about 130 B.t.u. per lb. which corresponds with a

velocity of some 2550 ft. per sec. With a proper allowance for nozzle friction, the velocity of the jet acting upon the wheel becomes about 2400 ft. per sec.

Taking a blade speed of 500 ft. per sec. and a nozzle angle of 20 deg., the direction in which the steam actually approaches the moving bucket and its velocity in that direction would be indicated by the direction and length of the line *bc* in Fig. 3, completing the triangle *abc*, of which the line *ac* (representing by its length and position the 2400 ft. velocity and the direction of the issuing jet) and the line *ab* (representing by its length and position the velocity and direction of movement of the bucket) are the other sides. The bucket changes the direction indicated by the arrow-head R_1 in Fig. 3, to that indicated by R_2. Reducing the length of this line in proportion to the loss of velocity by friction in the bucket and setting off *bd* equal in length to the bucket speed and drawn in the line of the bucket motion, the line *de*, indicating by its length and direction the velocity and direction of the steam as it issues from the blade, is found. Although the energy remaining in the jet is only as the square root of its velocity, so that if the velocity is halved three-quarters of the energy is taken out, the length of the line *de* shows that there is still enough velocity to make it worth while to redirect the jet upon the wheel.

This is done through the reversing chamber *B*, friction decreasing the absolute velocity to 1200 ft. per sec. and thus establishing an outlet angle of 15 deg. for the blade angle of 25 deg. Under these conditions the steam jet issues at right angles to the wheel with a small residual velocity of 250 ft. per sec., as shown in the lower left-hand diagram of Fig. 3, thus reducing the loss from this source to a minimum.

CASE II: Assuming now the pressure range from 165 lb. to 15 lb., both absolute. The steam is expanded from 165 to about 30 lb. in the first stage. The available heat would still be 138 B.t.u. and therefore the velocity developed remains as above. Then the high-pressure

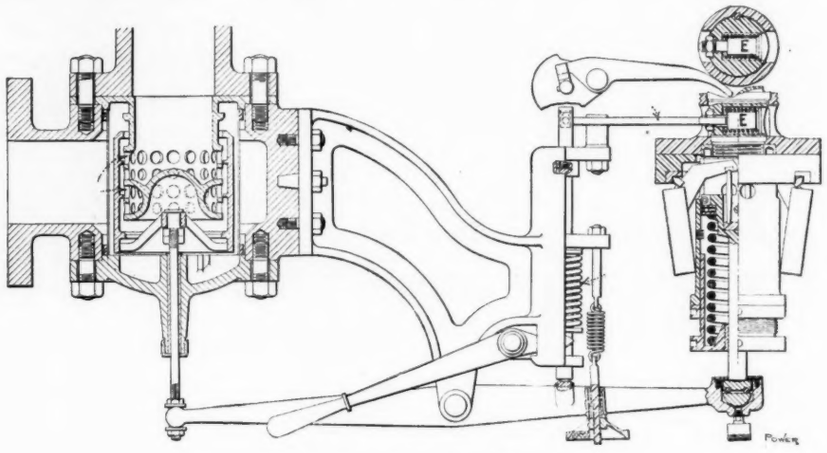


FIG. 8. THE ADMISSION VALVE

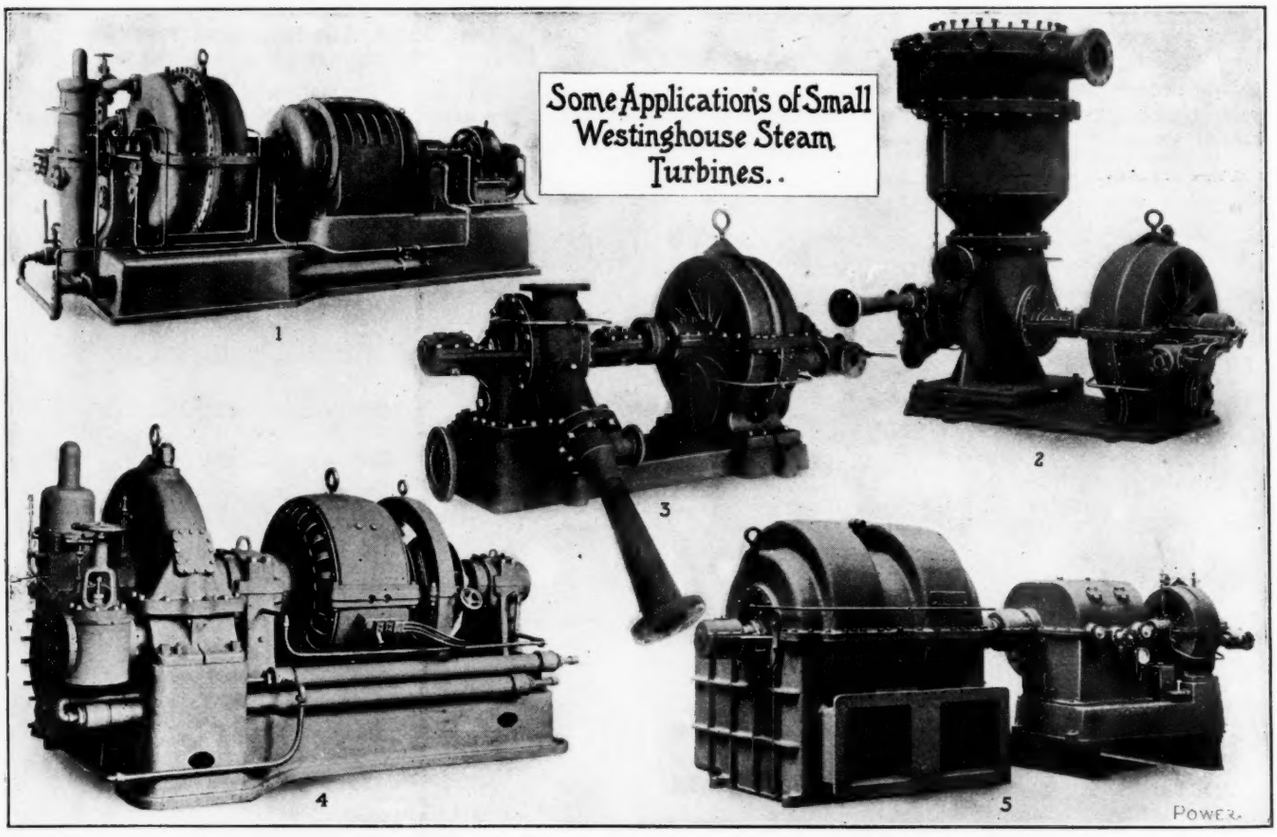


FIG. 9. 1. 100-KW. CONDENSING TURBINE DRIVING 60-CYCLE GENERATOR WITH DIRECT-CONNECTED EXCITER AT 3600 R.P.M.—2. LE BLANC CONDENSER, BOTH PUMPS DRIVEN BY TURBINE—3. AIR PUMP OF LE BLANC CONDENSER DIRECTLY CONNECTED TO TURBINE—4. A 100-KW. 125-VOLT DIRECT-CURRENT NON-CONDENSING SET—5. *A* 25,000-GAL.-PER-MIN. PUMP, TURBINE DRIVEN WITH REDUCTION GEAR

nozzles and reversing chambers *A* and *B* in both cases will be similar except that in Case II they will be of less width, due to the greater steam density. However, with the pressure of about 30 lb. at the end of the first stage, a velocity of some 1400 ft. per sec. may be developed by further expanding to 15 lb. absolute which is accomplished by the addition of the nozzle *C*, shown in the sketch illustrating Case II in Fig. 3. Had the initial pressure been higher, the velocity at *C* might have been sufficient to warrant the addition of another reversing chamber in the second stage. Depending upon the pressures and speeds used a third stage may be profitably employed. Numerous combinations are shown in Fig. 2.

Small turbines are naturally divided into two classes, those used for driving direct- or alternating-current generators, and power turbines such as are applied to blower or pump drive. A subdivision may be made, depending upon whether the service is condensing or noncondensing, although in small sizes the latter is the more common case.

Figs. 4 and 5 show sections of the noncondensing power and electric machines. While exactly similar in operating principle, there is some difference in detail. For power work, the shaft governor gives amply close regulation, but in the electric machines where the speed must be controlled within narrow limits over a wide range of load, the governor employed is the same as that used on the large Westinghouse turbines. But two bearings are used in the noncondensing electric units, one on each side of the generator, the turbine disk being overhung, as shown in Fig. 4. These bearings are lubricated by a closed system served by a pump on the governor shaft. The power-turbine bearings are ring-oiled. The construction of the glands for preventing the leakage of steam from the casing, is also different, the power turbines having free packing rings fitted in grooves turned in the hubs of the disks, whereas the electric machines have a hydraulic sealing device.

Fig. 6 shows the wheel of a noncondensing power or electric machine with one nozzle and reversing chamber clamped in the proper position relative to the wheel. As the photograph indicates, these chambers are bolted to the inside of the turbine casing which forms one wall of the steam passage.

The casing is split horizontally in both types, and the upper half may be removed without disturbing any connections. The lower half, which carries the nozzles and reversing chambers, is also split vertically in the plane of the wheel. Since the steam is expanded to approximately atmospheric pressure in the nozzles, the casing is designed for a moderate final pressure, any excess of which is prevented by a relief valve which is indicated in Fig. 4 and may be seen in the various photographs. The runners of the turbines, both electric and power, are of boiler-plate steel bolted to the hubs. Blades are formed with roots which set in a groove in the periphery of this disk, and are through-riveted. The material used for the blades is nickel-iron which is of high tensile strength, offers great resistance to corrosion and erosion, and has practically the same coefficient of expansion as the disk, thus avoiding strains due to unequal expansion when the casing is filled with steam.

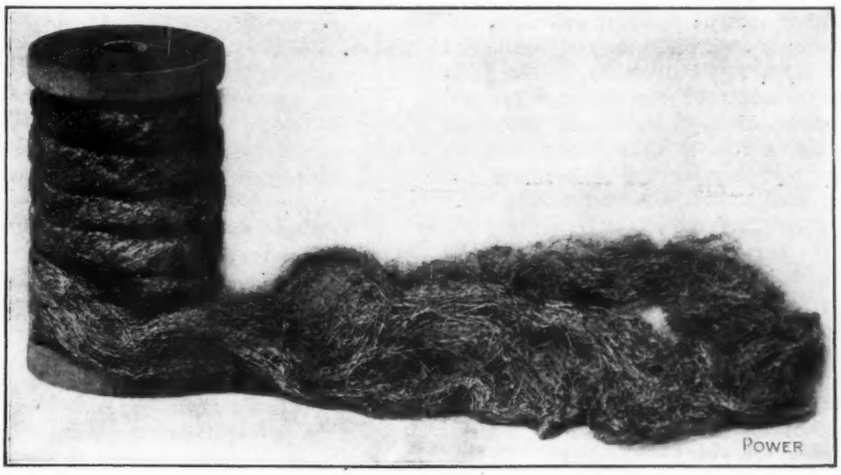
For condensing service, where steam volumes become great, a modified construction is necessary. The power turbines which run at slow speeds are more affected than the electric machines. Fig. 7 shows the runner of a 450-hp. condensing turbine to operate at 1000 r.p.m. The first-stage nozzle and reversing chamber of such a machine directs the steam upon the outer row of blades only. The second-stage nozzle makes use of the two outer rows, and the third and subsequent passes, if the latter are necessary, use all three rows of blades. In the electric machines the blade speed is such that except in the high-pressure stage no reversing chamber is necessary. During the rest of the expansion, there is a set of nozzles for each passage of the steam through the blades.

Fig. 8 shows the admission valve used on all the impulse turbines made by the Westinghouse Machine Co. The construction combines poppet and slide-valve characteristics. When closed, it is double-seated. After lifting from the seats approximately ⅛ in., the sleeve uncovers ports in the valve body which admit steam to the nozzles. This construction prevents scoring of the valve seat and consequent leakage when the turbine is operating at light loads. The connection of the valve stem with the governor linkage, and the operation of the safety overspeed stop are also evident from the engraving.

The first small turbines built by the Westinghouse Machine Co. were used to drive the discharge and air pumps of the Westinghouse-Leblanc jet condenser. There has been developed since then a complete line of turbine-driven powerhouse auxiliaries, including boiler-feed pumps, blowers, hotwell pumps and circulating pumps for surface condensers. Since the head against which circulating pumps for surface condensers usually discharge is relatively low, and the volume of water to be handled is large, the direct connection of a turbine to the pump is hardly possible on account of the slow speed involved. In several instances of large condenser installations, this difficulty has been overcome by the use of a reduction gear between the turbine and pump. In the group of photographs reproduced herewith is shown a 25,000-gal.-per-min. pump so driven, discharging against 13½ ft. total head. The pump runs at 300 r.p.m. and the turbine at 3000. The gear efficiency, due to the flexibility supported pinion shaft used, is 98 per cent. The group shows various other applications of the small turbine.

Ulco Lead Rope

This rope is made of extremely thin, long fibers of pure lead, and is principally used for jointing gas and water pipes where liquid metal could not be poured into the joint, owing to restrictions.



ULCO LEAD ROPE

When wrapped in cheesecloth or sheet rubber, lead rope makes an efficient and satisfactory metal packing, and can also be used on piston rods, valve stems for water, gas and steam.

When folded in cheesecloth or rubber and placed between flanges of high-pressure lines, lead ropes makes a strong gasket. There is no limit to size or thickness, and it can be used in any form or shape. The illustration shows a spool of Ulco rope, and the manner in which it can be separated. It is manufactured by the United Lead Co., 111 Broadway, New York City.

How Heat Flows into the Boiler Water

Henry Kreisinger and Walter T. Ray, of the Bureau of Mines, in their recent work entitled "The Transmission of Heat into Steam Boilers," published by the Bureau as its Bulletin 18, describe the passage of the heat from the furnace to the boiler as follows:

The accompanying sketch shows diagrammatically the modes of heat travel into boiler water.

The diagram shows the metal plate covered with a layer of soot on the gas side and with a layer of scale on the water side. Next to the soot layer and entangled in its recesses is a layer or film of motionless gas. This film of gas under ordinary conditions adheres so tightly to the soot or the metal that it may almost be considered a part of the solid plate. It is reasonable, therefore, to assume that the dry surface of the plate is situated somewhere within this film of gas, as indicated in the sketch. The wet surface of the heating plate is perhaps located on the inside of the boiler in a film of water and steam adhering to the layer of scale, or, if the boiler is clean, to the metal plate.

In any steam-generating apparatus the heat is produced in the furnace by the burning of fuel. As the process of combustion goes on the heat evolved is immediately absorbed, partly by heating of the fresh fuel but mainly by the gaseous products of combustion. The absorption of heat by these substances causes a rise in their temperature, so that by observing the temperature one can readily determine how much heat they contain.

The heat evolved and contained in the gaseous products of combustion is transferred through the gas-filled space and then transmitted through the heating plates into the boiler water. The process of transmission takes place in the three distinct modes shown in the sketch, each of which is governed by a definite law not applicable to the others. Before the heat reaches the body of the boiler water it changes its mode of travel at least twice. It is first imparted to the dry surface of the heating plate in two ways: (*a*) By radiation from the hot fuel bed, the furnace walls, and the luminous flames, and (*b*) by convection from the hot, moving gaseous products of combustion. When the heat reaches the dry surface it changes its mode of travel and passes through the soot, metal and scale to the wet surface purely by conduction. From the wet surface of the plate the heat is carried into the boiler water mainly by convection.

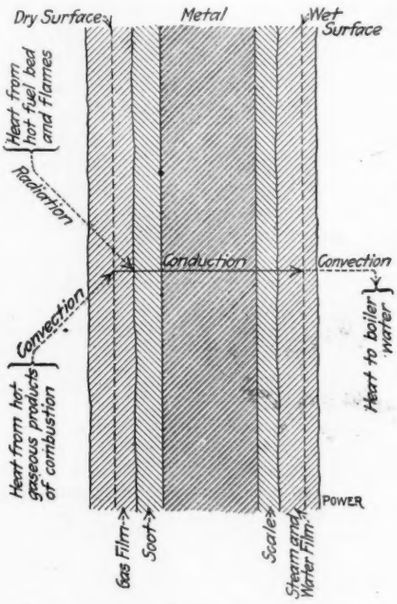
It may be well to define and explain briefly these modes of heat travel and the dry and the wet surfaces of the heating plate.

Radiation

Radiation is that mode of heat propagation by which heat passes through space from one body to another without any material agency; the two bodies do not come into contact directly nor does a third body or other bodies transmit the heat. In the case of a furnace and the heating plate of a boiler, the heat from the hot fuel bed and furnace walls flows by radiation through the space filled with gases, without heating the gases appreciably, directly into the boiler plate. The quantity of heat transmitted by radiation would not be lessened if the gases did not fill the space; in fact, it would be slightly greater if there were a vacuum between the fuel bed and the plate.

Convection

Convection of heat from one place to another always implies the motion of a



How Heat Flows into a Boiler

fluid receiving or giving up the heat. Thus, in the case of the hot gases and the boiler, the heat is transferred to the heating plate (from the gas) by small particles of the gas moving from the body of the gas to the plate and imparting their heat to the latter by their impact upon it. In other words, convection is a process of continuous interchange of position between cooled particles (molecules) of gas next to the heat-absorbing surface and hotter particles within the body of the gas. The quantity of heat imparted to a unit area of dry surface of the plate depends on the rate at which these particles of gas exchange positions.

Conduction

Conduction of heat is the process by which heat flows from a hotter body to a colder one, when the two bodies are in contact, or from a hotter to a colder part of the same body. Conduction implies no visible nor mechanical motion of different bodies or different parts of the same body, but implies a direct contact. In a boiler the heat passes by conduction from the particles of hot gas that touch the soot coating, into the soot, and then through the metal and the scale (if there happens to be any of the latter) into the particles of water next to the scale. These particles of water absorb heat until they change into small bubbles of steam which are carried away by the circulation of the water. Since the heat in these steam bubbles is taken from the heating plate through the water into the steam space by mechanical, or visible motion, it is said to travel by convection.

The particles of gas in contact with the soot coating of the tubes may form a gas film of appreciable thickness; then the heat near the outside of the film has to travel by conduction through part of the thickness of the film a longer distance than does the heat near to the soot coating. Thus the boundary along which convection ceases and conduction starts is not a well defined surface, but rather a gas layer of measurable thickness. In addition, gases are permeable to radiation, so that the heat traveling by radiation penetrates the gas film and enters the soot coating almost directly. In consequence, the dry surface of the heating plate may be defined as the thin layer of gas near, or at, the outside surface of the soot coating, where the heat ceases to travel by convection and starts to travel by conduction. For similar reasons the wet surface of the heating plate may be defined as the thin layer of steam and water, near or at the surface of the layer of scale away from the metal plate, where the heat ceases to travel by conduction and starts to travel by convection facilitated by the circulation of the water in the boiler.

The smallest central station in the United States is said to be in Sacramento, Neb., which has a grand total population of 12. However, Sacramento is progressive, as evidenced by the fact that it already has a central station. E. G. Anderson, proprietor of this embryo lighting company, operates a 3-hp. Fairbanks-Morse coal-oil engine having a total connected load of 30 tantalum lamps.—*Electrical World.*

The United States Navy's coaling station at Bradford, R. I., will shortly become one of the Navy's largest fuel-oil supply stations when a new tank, foundations for which are now being laid, is completed. It will have a capacity of 750,000 gal., which will give the station a total storage capacity of more than 1,500,000 gal. The oil will be mostly for the use of the torpedo-boat destroyers.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

Counter Electromotive Force

As explained in a lesson on the dynamo, the passage of a wire across a magnetic field causes an electromotive force to be generated in the wire. This is true of the wires on a motor armature as well as those of a dynamo. The electromotive force induced in the wires of a motor armature is in the opposite direction to the flow of the current which is supplied to the motor to drive it; the induced pressure is therefore called "counter electromotive force."

The counter electromotive force of a motor armature tends to prevent the passage of current through the windings, but it can never rise high enough to do so because that would deprive the armature of the power which drives it and thereby causes the generation of the counter electromotive force. The latter, however, is almost as great as the impressed electromotive force which forces the current through the armature; the difference between the two is the "drop" in the armature and its connections due to their electrical resistance.

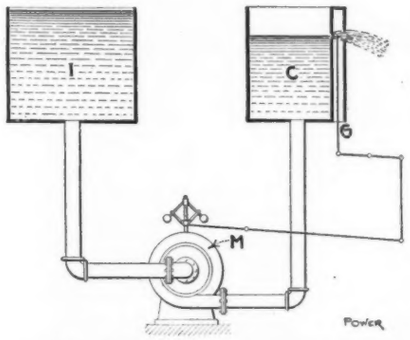


Fig. 107. Mechanical Analogue of Relations between Armature Electromotive Forces and Speed

For example, if the complete armature resistance of a 110-volt motor were ⅓ ohm and the armature current were 15 amperes, the drop due to resistance would be

$$\frac{1}{3} \times 15 = 5 \text{ volts}$$

The difference between this and the supply voltage would be the counter electromotive force, which, in the case assumed, would be 105 volts.

Fig. 107 illustrates crudely the principle of impressed and counter electromotive forces and motor speed. A water motor *M* is supplied from a tank *I* which is kept filled from an outside source. The discharge from the motor is delivered to another tank *C* and the level in this tank is supposed to be controlled by a gate *G*, the position of which is controlled by a flyball governor driven by the motor. The gate and its mechanism are supposed to be frictionless. When the motor is stationary, there is no water in the counter-pressure tank *C*, the gate being at its lowest position, opening the whole side of the tank. Therefore, when the water is first turned on, the motor is subjected to the full head of the tank *I*.

Suppose the motor to be running at normal speed and the main water level to be 110 in. above the outlet; if the resistance of the water passages be such as to require a head of 5 in. to force water through the system at such a rate as to keep the motor speed normal, then the governor would adjust the gate *G* to a level 105 in. above the motor outlet, giving the required difference of 5 in. between the impressed head in the tank *I* and the counter head in the tank *C*. Should the load on the motor be increased, the motor would slow down, thereby lowering the gate *G* and reducing the counter head until the difference between the two heads was sufficient to force the required water through. This is almost a complete mechanical counterpart of the behavior of an electric motor under the opposing influences of its counter electromotive force and the impressed electromotive force.

The counter electromotive force of a motor armature depends on the quantity of magnetic flux that passes between the field magnet and the armature core, the number of wires around the armature core, the way the wires are connected together and the speed at which the armature revolves. The way the wires are connected together is taken care of by stating the number of paths through the winding. Expressed as a formula:

$$\frac{Flux \times wires \times r.p.m.}{60 \times paths \times 100{,}000{,}000} = counter\ e.m.f.$$

In this formula, "flux" means the total magnetic lines of force passing between the pole faces and the armature core.

Speed

The counter electromotive force of a motor armature varies exactly with the speed, other things being fixed, but this is putting the cart before the horse in considering motors. The practical viewpoint is that the speed depends on how much counter electromotive force the armature is allowed to generate. Transposing the preceding formula will show that relation, thus:

$$\frac{6{,}000{,}000{,}000 \times paths \times counter\ e.m.f.}{Flux \times wires} = r.p.m.$$

The counter electromotive force being equal to the difference between the impressed electromotive force and the resistance drop in the armature, it can be easily calculated if the armature current and the resistance of the armature circuit be known: Thus:

Impressed electromotive force — (resistance × current) = Counter electromotive force

From this last formula it will be evident that the counter electromotive force of a motor cannot be constant unless the impressed electromotive force and the armature current are constant or else are varied oppositely (it being assumed that the armature resistance is constant, which is almost always approximately true). However, the "drop" in the armature circuit is usually small as compared with the impressed electromotive force; consequently, the speed of a "constant-potential" motor does not vary much if the impressed electromotive force is maintained steady.

For example, suppose that a 110-volt motor had an armature resistance of ⅓ ohm, its full load armature current were 15 amperes and the current at no load 1½ amperes. At full load the drop would be 5 volts, leaving 105 volts for the counter electromotive force to balance, and at no load it would be ½ volt, leaving 109½ volts to be balanced by counter electromotive force. If the speed at full load were 1500 r.p.m., the speed at no load would be 1564 r.p.m., assuming no change in the magnetic flux maintained by the field magnet.

Reference to the speed formula will show that changes in counter electromotive force can be compensated by making proportionate opposite changes in the field flux. For example, in the case just assumed, if the field flux at full load were 2,100,000 maxwells (lines of force), increasing this to 2,190,000 maxwells at no load would exactly compensate the change in counter electromotive force and maintain the speed constant. Perhaps working through the whole problem will make this clearer. Assume, therefore, the following values:

	No load	Full load
Armature wires	400	400
Armature paths	2	2
Armature resistance (ohms)	⅓	⅓
Armature current (amp.)	1½	15
Armature drop (res. × current)	½	5
Impressed e.m.f.	110	110
Counter e.m.f.	109½	105
Normal magnetic flux (maxwells)	2,100,000	2,100,000

Substituting these values in the formula for speed gives the following comparison:

$$R.p.m.\ at\ full\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 105}{2{,}100{,}000 \times 400} =$$

$$R.p.m.\ at\ no\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 109\frac{1}{2}}{2{,}100{,}000 \times 400} = 1564$$

Increasing the flux at no load to 2,190,000 maxwells:

$$R.p.m.\ at\ no\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 109\frac{1}{2}}{2{,}190{,}000 \times 400} = 1500$$

In special cases, the field flux is varied automatically with variations in the armature current of a constant-potential motor in order to keep the speed constant. The large majority of such machines, however, are operated with a steady magnetic field, because the speed variation is not great enough to make it worth while to complicate the construction of the machine in order to avoid it.

Power

The mechanical power of an electric motor is commonly expressed in the familiar terms of "horsepower"; one horsepower being the equivalent of 33,000 foot-pounds expended per minute, as in all other machines. One horsepower is also equivalent to 746 watts* as explained in an early lesson on units. The mechanical power delivered by a motor is equal to the power which it receives from the supply circuit, minus the power absorbed by the machine itself. Expressed differently, the delivered power is equal to the applied power multiplied by the efficiency of the machine. Thus:

$$Applied\ power \times efficiency = Delivered\ power$$

The power must, of course, be expressed in the same units both times, and the efficiency must be written as a fraction or a decimal. For example, if the applied power be 1865 watts and the efficiency 80 per cent., the delivered power will be

$$1865 \times 0.80 = 1492\ watts$$

and this is equivalent to

$$\frac{1492}{746} = 2\ hp.$$

There is also a relation between torque and speed and horsepower. Multiplying the torque by the number of revolutions per minute and by 2π (6.2832) gives the foot-pounds of work done per minute; dividing this by 33,000 gives the horsepower. Thus:

$$\frac{6.2832 \times r.p.m. \times torque}{33{,}000} = hp.$$

This can be proved by going over the data of an actual machine and checking through the electrical and mechanical figures. These figures are as follows:

Armature diameter (inches)	4
Full load r.p.m.	1,500
Full load armature current (amp.)	15
Number of armature paths	2
Current in each wire (amp.)	7½
Number of armature wires	400
Total magnetic flux, maxwells	2,100,000
Delivered horsepower	2
Power absorbed in driving armature (hp.)	0.11
Total full-load hp. developed	2.11

According to the formula previously given, the torque was

$$\frac{2{,}100{,}000 \times 400 \times 7\frac{1}{2}}{852{,}230{,}000} = 7.39\ lb.\text{-}ft.$$

Transposing the last formula:

$$\frac{33{,}000 \times hp.}{6.2832 \times r.p.m.} = torque$$

and filling in the known values for horsepower and speed:

$$\frac{33{,}000 \times 2.11}{6.2832 \times 1500} = 7.39\ lb.\text{-}ft.$$

The interested student will find it worth while to check back even farther, verifying the figures for horsepower. The input of the armature was

110 *volts* × 15 *amperes* = 1650 *watts*

The resistance of the armature circuit was ⅓ ohm and the drop, therefore, was ⅓ × 15 = 5 volts. This makes the power lost in overcoming the electrical resistance 5 × 15 = 75 watts, leaving 1650 — 75 = 1575 watts for driving the armature and doing useful work at the pulley. The power required to turn the armature at 1500 r.p.m. was found by test to be 83 watts. Subtracting this from the 1575 watts available leaves 1492 watts for work outside the motor; 1492 watts equals 2 hp.

The mechanical power developed by a motor armature, when expressed in watts, is equal to the counter electromotive force multiplied by the armature current. This includes the power wasted by friction and core losses as well as the useful power delivered at the pulley. In the case just considered, for example, the counter electromotive force was 105 volts and the armature current 15 amperes; the product of these (105 × 15) is 1575 watts, which has just been shown to cover 83 watts lost by friction and armature core effects and 1492 watts delivered as useful work.

Efficiency

There are several kinds of efficiency, but unless some particular kind is named, the efficiency of a motor is commonly intended to mean the ratio of output to input, thus:

$$\frac{Output}{Input} = efficiency$$

The output, as previously stated, is equal to the input minus the losses in the machine. Since the input and losses are almost always measured or computed in watts or kilowatts, it is advisable, when discussing efficiency, to express the output in watts, to avoid confusion. This is done by multiplying the output in horsepower by 746.

There are four sources of loss in a motor: the resistance of the field winding, the resistance of the complete armature circuit, the friction of the moving surfaces (the shaft in its bearings, the commutator under its brushes and the armature surfaces sweeping through the air) and hysteresis and eddy currents in the armature core. These were all fully explained in a previous lesson on the dynamo. Expressing the losses in watts, they are simply added together to get the total losses for subtraction from the input.

Taking as an example the motor previously discussed, the following values are obtained:

Full load conditions	Amperes	Volts	Watts
Armature winding	15	5	75
Field winding	1.95	110	215
Armature core and friction			83
Total losses			373

The input from the line was 16.95 amperes at 110 volts, making 1865 watts. Subtracting 373 from this leaves 1492

TABLE 6. AMPERES PER HORSEPOWER

Efficiency %	110 V.	115 V.	120 V.	220 V.	230 V.	240 V.	250 V.	500 V.	550 V.	575 V.
75	9.04	8.65	8.29	4.52	4.32	4.15	3.98	1.99	1.81	1.73
76	8.92	8.54	8.18	4.46	4.27	4.09	3.92	1.96	1.78	1.71
77	8.81	8.42	8.08	4.40	4.21	4.04	3.88	1.94	1.76	1.68
78	8.70	8.32	7.97	4.35	4.16	3.99	3.82	1.91	1.74	1.66
79	8.58	8.21	7.87	4.29	4.11	3.93	3.78	1.89	1.72	1.64
80	8.48	8.11	7.77	4.24	4.05	3.89	3.74	1.87	1.70	1.62
81	8.37	8.01	7.67	4.19	4.00	3.84	3.68	1.84	1.67	1.60
82	8.27	7.91	7.58	4.14	3.96	3.79	3.64	1.82	1.65	1.58
83	8.17	7.82	7.49	4.08	3.91	3.74	3.60	1.80	1.63	1.56
84	8.08	7.72	7.40	4.04	3.86	3.70	3.56	1.78	1.61	1.54
85	7.98	7.63	7.31	3.99	3.82	3.66	3.52	1.76	1.60	1.53
86	7.89	7.54	7.23	3.94	3.77	3.61	3.48	1.74	1.58	1.51
87	7.79	7.46	7.15	3.90	3.73	3.57	3.44	1.72	1.56	1.49
88	7.71	7.37	7.06	3.85	3.69	3.53	3.40	1.70	1.54	1.47
89	7.62	7.29	6.99	3.81	3.64	3.49	3.36	1.68	1.52	1.46
90	7.54	7.21	6.91	3.77	3.60	3.45	3.32	1.66	1.51	1.44
91	7.45	7.13	6.83	3.73	3.56	3.42	3.28	1.64	1.49	1.43
92	7.37	7.05	6.76	3.69	3.53	3.38	3.24	1.62	1.47	1.41
93	7.29	6.97	6.68	3.65	3.49	3.34	3.20	1.60	1.46	1.39
94	7.22	6.90	6.61	3.61	3.45	3.31	3.18	1.59	1.44	1.38
95	7.14	6.83	6.54	3.57	3.41	3.27	3.14	1.57	1.43	1.37

*The exact equivalent is 745.65 watts, but 746 is commonly used.

watts for useful work. The efficiency, therefore, was

$$\frac{1492}{1865} = 0.80$$

or 80 per cent.

As motors are operated on circuits of standard voltages, it is usual to take the input in amperes instead of watts, in considering a motor's performance. Table 6 gives the number of amperes required at each of the standard voltages to enable a motor to deliver one horsepower, at different efficiencies ranging from 75 to 95 per cent. The table will be found convenient for ascertaining how much current a given motor will require, the efficiency being known or estimated, and also for ascertaining the efficiency of a motor of a given rating in amperes and horsepower.

Example I. What is the full-load current of a 20-hp. motor of 88 per cent. efficiency, built to run on 240 volts?

Starting at 88 in the efficiency column of the table and following that line over to the 240-volt column, the number found there is 3.53 (amperes per horsepower). Multiplying this by 20 gives 70.6 amperes as the full-load current.

Example II. A 220-volt motor has a nameplate stamped "15 hp." and "60 amperes." What is its efficiency?

If the current at 15 hp. is 60 amperes, the current per horsepower will be

$$\frac{60}{15} = 4 \text{ amperes}$$

In the 220-volt column, the number nearest to 4 is 3.99 and in the line with this, in the efficiency column, is 85; the efficiency at rated load, therefore, is 85 per cent.

Running 60-Cycle Alternator at 50 Cycles

In answer to an inquiry by one of our readers as to how much is lost in the efficiency of a 2300-volt alternator if designed for 60 cycles but run at 50 cycles, we offer the following comment:

The efficiency of a generator is the ratio of the power delivered at the terminals to this amount plus the internal losses. Thus, if the output in watts is designated by P, the armature copper loss by L_A, the field copper loss by L_F the hysteresis loss by L_H, the eddy current loss by L_E, and the loss due to friction and air resistance by L_W, the efficiency can be expressed as follows:

$$\text{Efficiency} = \frac{P}{P + L_A + L_F + L_H + L_E + L_W}$$

When an alternator designed for a frequency of 60 cycles, is operated at 50 cycles, several quantities in the foregoing formula are increased. A fundamental principle in generator design is that the voltage depends upon the density of field flux, the speed of the moving element and the number of conductors in series on the armature. It therefore follows that in lowering the speed such that the frequency is reduced from 60 to 50 cycles, the voltage, if maintained constant, must be upheld by strengthening the field flux. This necessitates a heavier current in the field winding, with a consequent increase in the field copper loss. The stronger field also increases the iron losses in the armature and field frame. The variation in the friction and air resistance will be so small as to be practically negligible. If the increased values of these losses are substituted in the formula, it will be seen that the efficiency will be materially reduced.

In order to accurately determine the efficiency mathematically, for a 60-cycle alternator when operated at a frequency of 50 cycles, detailed information relative to the design of the machine must first be secured. The diameter of the air space must be known, together with the number and size of the field poles, the dimensions and construction of the armature core, the qualities of the iron used, the type of armature winding, the excitation voltage and many other features, all of which are involved in the calculations of the various copper, iron and mechanical losses. None of these quantities are stated in the question; hence owing to the wide variety in alternator sizes, designs and characteristics, and the consequent practical impossibility of citing any efficiency values at various frequencies which would accurately apply to alternators in general, no reliable definite value for the desired loss can be given.

CORRESPONDENCE

What Reversed the Motor?

In reply to the question, appearing in POWER for Oct. 29, "What Reversed the Motor?" I wish to submit the following:

The motor reversed its direction of rotation because its field magnetism was reversed by inserting the lamp in place of the resistance coil in the field circuit.

Assuming the ¼-hp. motor to have an efficiency of 60 per cent., the normal watts consumed per hour would be

$$\frac{746 \times 100}{4 \times 60} = 311 \text{ watts}$$

With the motor operating as a series machine under full load the current in the field coils would be

$$\frac{311}{110} = 2.83 \text{ amp.}$$

Thus 2.83 amp. is the normal magnetizing current. When the 50-watt lamp was inserted in place of the resistance coil the current in the field winding was reduced to $\frac{50}{110} = 0.45$ amp. or less, as the voltage across the lamp was reduced because of the resistance drop in the field coils. In this way the lamp reduced the field current to less than ⅙ its normal value. Because the magnetism in the field poles increases very slowly until a certain current value is reached, the magnetic field produced was very weak.

The armature winding of the motor may be divided into two parts, one consisting of coils which produce a cross-magnetizing field only and the other of coils which produce a demagnetizing field only. The more the brushes are shifted from the axis of symmetry the greater the proportion of armature coils that tend to produce a demagnetizing field. This demagnetizing effect of the armature coils on the field destroyed the magnetism of the fields and magnetized them in the opposite direction.

Thus when either the field magnetism or the armature current are reversed in a motor its direction of rotation is reversed.

AMOS L. UTZ.

St. Joseph, Mo.

Mr. Boone's trouble is very similar to trouble I experienced with a 15-hp., 112-volt, shunt-wound motor, driving a circular saw. The motor was about 200 ft. from the generator and received its current over a No. 6 B. & S. wire. On light load it would operate satisfactorily; but with a heavy load the motor would stop, and start up in the reverse direction running a little above normal speed, with no sparking at the brushes.

A voltmeter showed 20 volts across the motor terminals when the armature was stalled, and would increase a few volts when the armature was running backward. By shifting the brushes forward slightly the machine would start in the proper direction, after stalling. I came to the conclusion that the polarity of the field was completely reversed by the effect of the armature current, which would account for the reversed rotation; also that the motor could not reverse its field again, since any increase of voltage at the field terminals would weaken the field caused by the armature current, which would increase the armature current, and drop in the line, lowering the voltage at the motor terminals again. I think Mr. Boone can trace his trouble to a weak field. By placing a lamp in series, he weakened the field much below normal, causing the armature current to be above normal, with a corresponding increase in the armature reaction, and with the large current in starting the already weak field was reversed.

ORAL CHANDLER.

Ft. Mansfield, R. I.

The first coal mine worked in the Southern States was opened in the Richmond basin, Virginia, in 1822. Soon after, mines were opened in Maryland, Kentucky, Alabama, Tennessee and Georgia.—*The Tradesman.*

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

A System of Gas Engine Governing *

By George S. Cooper

Among the problems that have arisen in connection with the widespread adoption of the internal-combustion engine, few have led to a wider variety of design than the governing problem.

As compared with steam-engine governing, the number of possible methods is much greater. The steam engine operates on a medium of practically unvarying characteristics as to pressure, temperature, and chemical composition, prepared in a separate piece of apparatus. Thus there is left to the engine's governing mechanism only the determining of the quantity of the medium to be admitted to the cylinder at each stroke. Since nearly all steam engines making any pretense to economy operate on the automatic cutoff principle, the duty of the steam-engine designer is to provide a mechanism which will cut off the admission at such a time that the mean effective pressure will be proportional to the power demand, thus holding the speed of the engine constant within 1 or 2 per cent.

In contrast to this simple requirement a much more complex function devolves upon the internal-combustion engine designer. The engine, if it be of the four-stroke-cycle type, must receive the fuel in a liquid or gaseous form, mix it thoroughly with the proper quantity of air to form a combustible mixture, admit this mixture to the engine cylinder during the suction stroke, compress it into the clearance space during the next stroke, ignite the charge and expand the products of combustion during a working stroke, and, finally, during the exhaust stroke, expel the spent gases from the cylinder, the whole cycle occupying two revolutions. In the two-stroke-cycle engine these events take place during one revolution consisting of a compression and a working stroke, the fresh charge being forced in and the exhaust gas expelled either by a separate pump, in double-acting engines, or by utilizing the crank end of the main cylinder or the crank case as a pump in single-acting engines.

As in the steam engine, the mean effective pressure must be graduated to the load driven if the speed is to be maintained constant. This may be done in a number of ways.

Where the so called "hit-and-miss" method is employed, a constant volume of mixture having a constant fuel and air ratio is admitted at each suction stroke until the speed rises above normal, when the governor acts to prevent the admission of fuel during one or more cycles until the speed drops slightly below normal; then the admission of fuel is once more resumed. This method has proved quite satisfactory for small engines, especially those of the single-cylinder, single-acting type, but for medium and large multi-cylinder machines more complete methods have been sought.

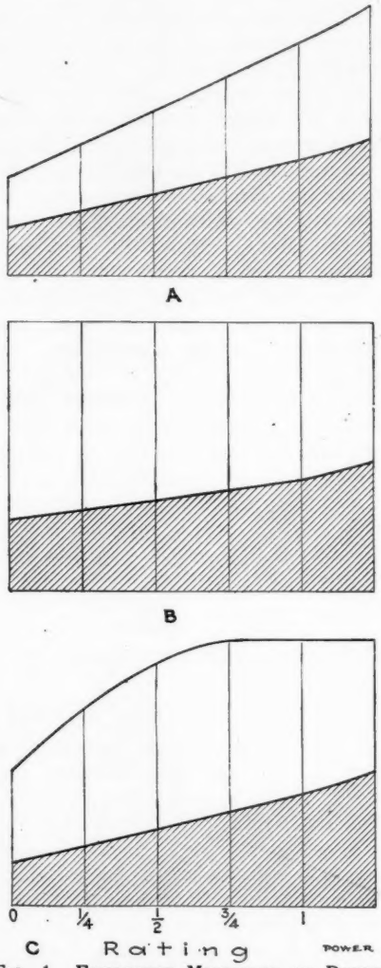


Fig. 1. Explosive Mixtures at Different Loads for Different Governing Systems

One of the most widely adopted governing methods is the "throttling" system. In this the engine receives a charge of constant fuel and air ratio at every suction stroke but the quantity of this charge is varied to suit the load on the engine by passing it through a throttling valve, the opening of which is controlled by the governor. Diagram A, Fig. 1, shows graphically how the quantity of the mixture is varied throughout the load range. The shaded portion of the diagram represents the volume of gas and the unshaded portion the volume of air used. Ordinates therefore represent the total volume of the explosive charge. Since the compression pressure is proportional to the volume of the charge admitted, it follows that the ordinates also represent the compression pressure, and their varying heights from no load to maximum show how the compression varies with the load.

The simplicity of the throttling method commends it but it is open to the objection that it involves a greatly reduced compression on light loads together with a loss of work due to wiredrawing the mixture on the suction stroke.

A third method is illustrated in diagram B. Here all loads from the lightest to the heaviest are carried by merely changing the ratio of gas to air. The total mixture volume does not vary and therefore the compression is constant throughout the load range. This method satisfies the theoretical requirements of full compression on light loads but involves the use, on light loads, of very lean mixtures which, due to the great excess of air present, burn with poor efficiency and frequently fail to ignite; consequently this method is open to much objection from the standpoint of fuel economy.

Diagram C illustrates a combination of the two latter methods which makes it possible to maintain economical compression on light loads without weakening the mixture to such an inefficient point. In this system all loads from maximum down to about 60 per cent. of the rated load are handled by simply varying the ratio of gas to air, the compression remaining constant. Loads less than about 60 per cent. of the rated capacity are handled by varying both the quality and the volume of the charge. By means of this combined action, the lightest load may be carried without impoverishing the charge to the extent involved in the plain quality method, and without reducing the compression to so low a value as in the throttling method.

The means by which this is accom-

*From a paper read before the Ohio Society of Mechanical, Electrical and Steam Engineers at Akron, Ohio, Nov. 21, 1912.

plished are shown in Fig. 2, which is a cross-section through the valves and combustion chamber of the Buckeye four-stroke-cycle gas engine. Exhaust valve *E* and the inlet valve directly above are operated by lifters and wiper cams driven from a single eccentric on the layshaft.

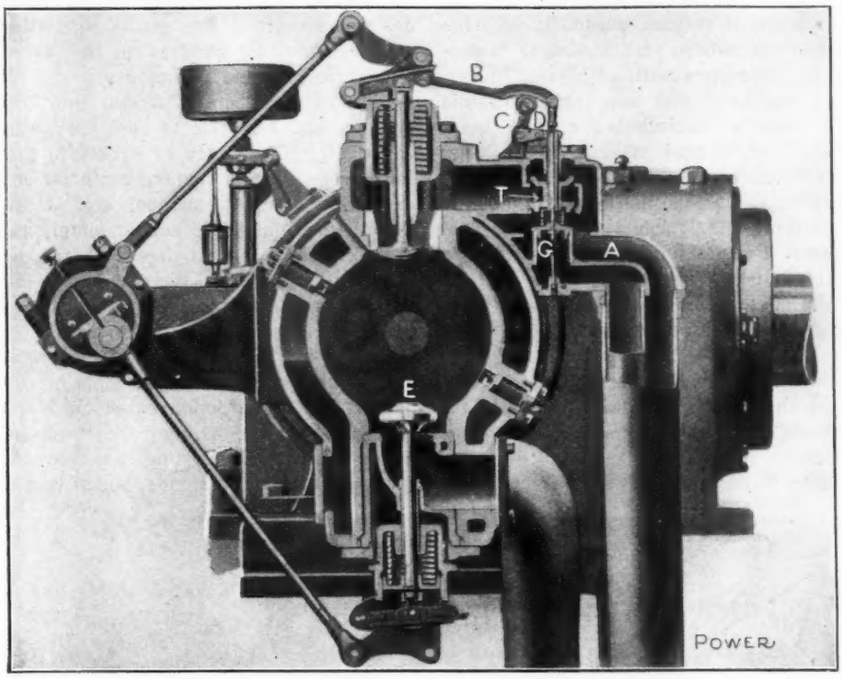


FIG. 2. CROSS-SECTION THROUGH COMBUSTION CHAMBER OF BUCKEYE FOUR-STROKE-CYCLE GAS ENGINE

Each combustion chamber of this engine is supplied with fuel by a separate mixing chamber. The gas enters the lower compartment of each mixing chamber through a small pipe inclosed within the air pipe. A gas valve *G* prevents the gas and air from mixing at all times except during the suction stroke. Valve *G* derives its motion from the main inlet-valve lifter through the segment lever *B* which fulcrums on a roller carried by lever *C*. This lever is under control of the governor with the result that, while the gas valve lifts at every suction stroke, the extent of its lift is variable, being greater at heavy loads and less on light loads. Surrounding the gas-valve stem is a tubular stem attached to the balanced throttling valve *T*. This valve is also governor-controlled through lever *D*. Note, however, that while the gas valve opens and closes at every suction stroke, the lift varying with the load, the throttling valve is held stationary as long as the load is constant and moves to a new position only when the load varies. At all loads above 60 per cent. of the rating this valve is held so high above its seat that the mixture is not throttled appreciably, the governing being done entirely by the variable lift of the gas valve. As the load falls off, the increased speed causes the governor weights to expand and move the fulcrum roller under the segment lever to a position where the gas valve will have a decreased lift, and at the same time the action of the governor reduces the opening of the throttling valve. All loads from maximum down to about 60 per cent. are handled purely by varying the mixture, the total volume of mixture and, consequently, the compression remaining unchanged during this range. At about two-thirds load the throttling valve begins to reduce the volume of mixture admitted to the cylinder so that all lighter loads are carried by weakening the charge and using less of it.

However, since the fulcrum roller has a relatively small movement between 60 per cent. and friction load, it is evident that the throttling effect strongly predominates during the lower part of the load range, that is, the quality of the charge varies but little on the lighter loads, governing being accomplished mainly by changing the volume of the charge. Since lean fuel mixtures burn more slowly than rich ones it has been found advisable to vary the ignition timing as the load on the engine varies. The governor does this automatically by advancing the ignition timer on light loads and retarding it on heavy ones.

This governing mechanism may be adjusted to plain quality governing, plain throttling governing, or any combination of the two according to the kind of gas used. With a view to determining the relative fuel economy of these three systems the writer ran a series of tests on a 150-hp. single-acting tandem engine equipped as described. Fig. 3 shows the results of these tests. The economical results obtained on maximum load are the same in all three systems, as is to be expected when it is recalled that, regardless of the system of governing employed, ultimate capacity is obtained by admitting at each suction stroke a cylinderful of mixture having that proportion of gas and air which will give the greatest mean effective pressure. The greatest differences appear on the light loads. The plain throttling system gives much better results than the quality system, while the combination method shows better economy than either of the others. Fig. 3 does not necessarily represent the best results obtainable under either system. It simply shows comparative results with the various governing methods, all other conditions being equal.

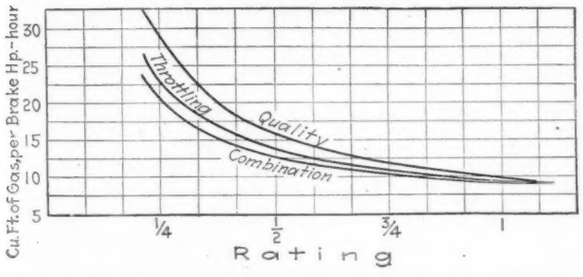


FIG. 3. RELATIVE FUEL FOR DIFFERENT GOVERNING SYSTEMS

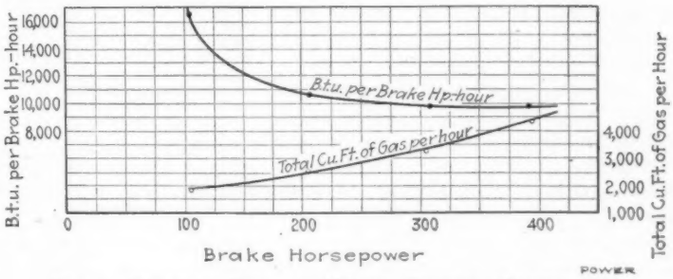


FIG. 4. ECONOMY TEST ON 400-HP. ENGINE, COMBINATION GOVERNED, USING NATURAL GAS

The combination method possesses certain operating advantages worth mentioning. First, owing to the fact that the governor controls the gas-valve lift, the quality of the explosive mixture is to an extent self-regulating; for example, should the pressure in the gas line drop, the action of the governor would

increase the lift of the valve, thus restoring the quality of the explosive mixture to normal. A further advantage is that maximum richness of mixture is employed only when the engine is working at its ultimate capacity. Over-rich fuel mixtures are wasteful because a portion of the gas is rejected unburned owing to the lack of oxygen.

The combination system, owing to its flexibility of adjustment, is readily adapted to any power gas in general use.

Natural gas and carbureted water gas have a range of explosiveness relatively small as compared with producer and blast-furnace gas. That is, the ratio of gas to air when using the first named gases must be kept nearer a constant figure than when using the last named. Therefore, engines using natural or illuminating gas are adjusted with the throttling effect predominating. When using producer and blast-furnace gas the quality may vary more widely; hence governing is accomplished by varying the gas and air ratio through the greater part of the load range and throttling is resorted to on the lightest loads only. Fig. 4 shows the results of an economy test of a 400-hp. engine, combination governed and operating on natural gas. Attention is called to the flatness of the heat-consumption curve between rated load and half load.

It is frequently stated that internal-combustion engines show their best fuel economy when operating at or near their maximum capacity. This is correct when the heat consumption is expressed in terms of net or brake horsepower; but the popular conception seems to be that this characteristic results alone from a decided falling off in combustion efficiency at all load conditions less than maximum, the conclusion being that the permissible range of compression pressures and mixture ratios is very narrow. The principal arguments favoring the "hit-and-miss" governing system are based on this hypothesis. When, however, the heat consumption is reduced to terms of indicated horsepower instead of brake horsepower, it is seen that the fuel consumption per indicated horsepower is almost constant from maximum down to about one-third load, and that the apparently high heat consumption on moderate loads is due not so much to a falling off in combustion efficiency as to the decreased mechanical efficiency resulting from the fact that engine friction does not decrease in proportion to the decreased load but remains nearly constant at all loads. For example, when the engine is working at less than about 20 per cent. of its rating, the power necessary to overcome friction is actually greater than the net delivered horsepower and, therefore, the heat consumption per brake horsepower is more than twice the consumption per indicated horsepower.

A Reversible Marine Oil Engine

By Edwin Lundgren

Recent years have brought a remarkable development of the oil engine, particularly for marine purposes, and for this class of service, especially in large sizes, reversibility is absolutely necessary. Propellers with adjustable blades, reversing gears and couplings, hydraulic and electric transmission can be used, but are really only makeshift; the large marine oil engine should be directly reversible just as the marine steam engine.

Viewing the achievements of the notable manufacturers of oil engines in Europe, it will be found that most of them have adopted the two-stroke-cycle engine for ship propulsion. Among these are Sulzer Bros. in Switzerland, Maschinenfabrik Augsburg-Nürnberg in Germany, Fiat in Italy, Atkiebolaget Diesels Motoren in Sweden and many others.

This is largely because it is easier to reverse a two-stroke-cycle engine than a four-stroke-cycle machine; the only thing that has to be different for running in the opposite direction being the time of fuel injection. The problem of reversing then is to cut off the fuel supply, stop the engine, and start it in the opposite direction by means of compressed air; then have the fuel injected at a different time. This is comparatively simple in a two-stroke-cycle engine. Still this cycle has some decided disadvantages. Admitting the mechanical simplicity attending it due to the absence of valves and valve-gear, and the lighter weight, it must be said that the fuel economy of the four-stroke cycle is not reached by the two-stroke-cycle engine. The latter will at least use 5 to 10 per cent. more fuel and more cooling water.

Of still greater importance, for a Diesel engine in particular, is the fact that it is much more difficult to keep the fuel-injection nozzle of the two-stroke-cycle engine free from carbon. This is due to the greater heat prevailing in the cylinder where there is one explosion during every revolution, whereas in the four-stroke-cycle engine during every second revolution fresh air is drawn into the cylinder and has time to cool the walls effectively. The order of events in the four-stroke-cycle is suction, compression, explosion, exhaust, suction, etc. It is evident that this cycle cannot simply be reversed, but that it is necessary to change the drive of the valve-gear or the valve-gear itself in order to obtain the proper succession of working events.

One of the best solutions of the problem which meets all requirements of reliability is a construction based on Weyland's patents and built by the Nordiska Motorverkstad in Finspong, Sweden. A general view of this engine, which is not a Diesel but of the low-pressure type, is shown in Fig. 1. The valves are actuated by a camshaft which can be shifted in axial direction to bring into play one of the two sets of cams—one for forward motion, the other for reverse direction.

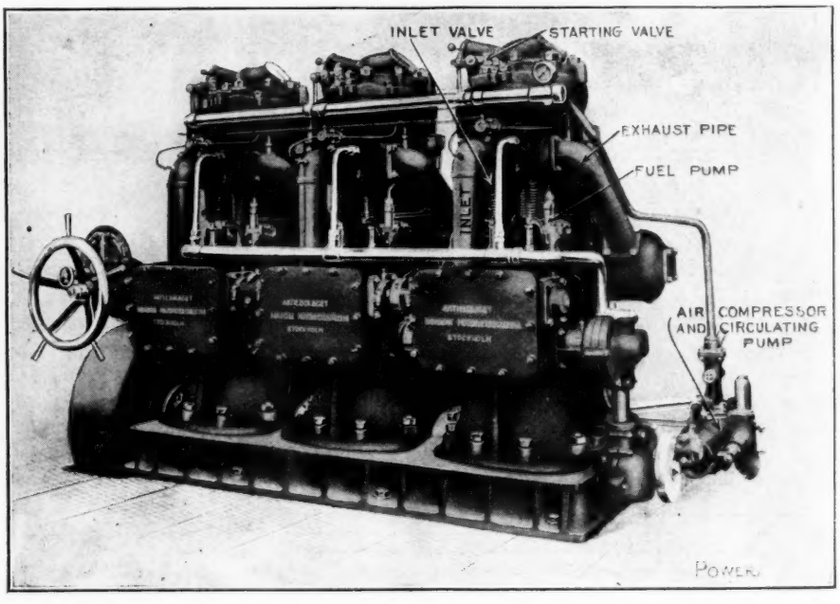


Fig. 1. General View of Engine

Besides this regular camshaft, driven by two pairs of spiral gears, there is another shaft, the so called operating shaft, which carries a number of cams, one each for the inlet, exhaust, fuel-injection and starting valves. The valve stem of each valve itself is lifted by means of a lever, the stationary fulcrum of which is formed by the edge pressing against the corresponding cam on the operating shaft by a spring; see Fig. 2.

It is apparent that by turning the op-

erating shaft with the cams, this edge is lowered or raised, thereby bringing the roller on the valve lever more or less within reach of the cam on the camshaft.

Referring to Fig. 2, *A* represents the fuel cam, *B* the starting-valve cam and *C* the exhaust cam, the lines of the inlet cam coinciding partly with those of the exhaust cam and partly with those of the fuel cam. For starting the engine the operating shaft is turned in such a position that cam *B* points downward. When the cam has been placed in this position the fuel valve is idle, while the compressed-air valve is actuated. As soon as the engine has gained sufficient momentum, the operating shaft is turned so that the lever for the fuel injection is pressed down against the camshaft bringing the fuel pump into action, while at the same time the lever for the air valve is allowed to recede toward the center of the operating shaft and thus is put out of action.

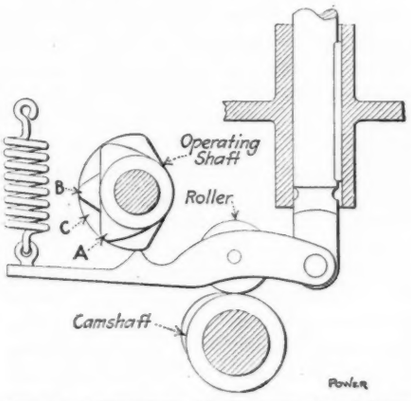


FIG. 2. OPERATING CAMS AND CAMSHAFT

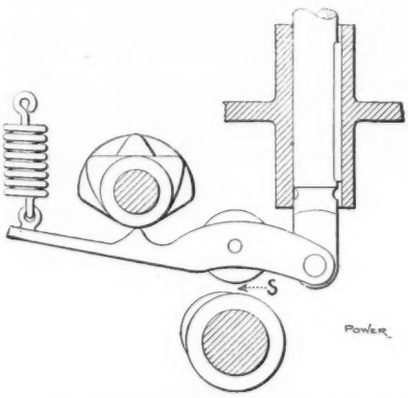


FIG. 3. NEUTRAL POSITION OF OPERATING SHAFT

The engine is then running in the direction wanted, and also at the speed wanted, as the lift of the fuel pump can be varied as desired. To reverse the engine the operating shaft has first to be turned into the position indicated in Fig. 3, where all the valve levers are clear of the cams on the camshaft and consequently all the valves remain closed. The camshaft is now shifted axially so that the cams for reverse direction are placed under the valve levers.

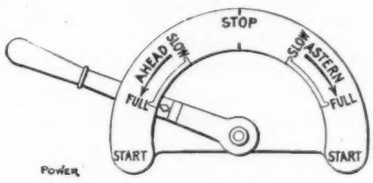


FIG. 4. CONTROL LEVER

The shifting motion is also derived from the turning of the operating shaft by means of an arm connecting the two shafts with bushed bearings on either shaft and provided with pins, one guided in a circular groove on the camshaft, the other in a cam-shaped groove on the operating shaft. The operating shaft is then turned further so that the engine is given a start by compressed air in the reverse direction. After a few revolutions the starting valves are cut out and the fuel pumps are put into action in precisely the same manner as described for running in the other direction.

All these operations are performed by throwing a single lever which actuates the operating shaft through bevel gears. Also the speed is under absolute control of this one lever. Fig. 4 represents the dial for different positions of the lever. The reversing can be done in a few seconds, and very little compressed air at 60 lb. pressure is required.

There are a number of ships equipped with these engines, particularly in Russian waters which give entire satisfaction. This reversing principle has also been applied to engines of the strictly Diesel type.

The Gas Engine as an Economical Power Producer*

Gas engines have been very greatly improved in design and construction during the last few years and undoubtedly offer the most economical means of producing power. The heat consumption of large gas engines in practical working is about 10,000 B.t.u. at normal full load and about 9500 B.t.u. at the maximum overload. The net heat value of blast-furnace gas varies with the duty of the furnace and the character of the fuel used, but in general varies between 90 and 110 B.t.u., or, say, 100 B.t.u. per cu.ft. as an average. Therefore, the consumption of blast-furnace gas per brake horsepower per hour would be approximately 100 cu.ft.

With coke-oven gas the heat consumption is the same. The heat value of coke-oven gas varies considerably, but is generally in the neighborhood of 450 to 500 B.t.u., occasionally less. Taking 450 B.t.u. as the usual figure, the consumption per brake horsepower per hour at normal full load is about 22½ cu.ft.

The exhaust gases from gas engines can also be used for generating steam, and when an engine is developing something like its full load, from 2 to 2¼ lb. of steam per brake horsepower at 60 lb. pressure is regularly generated, and this steam can be utilized for driving auxiliaries, heating and other purposes.

As regards the power available from blast-furnace and coke-oven plants, the following figures are interesting:

The calorific value and volume of gases evolved by a blast furnace depend upon the character of the furnace burden, and to some extent upon the method of driving, but, as an average figure (Northeast Coast practice) the gas evolved per ton of pig iron produced is about 160,000 cu.ft., measured at atmospheric temperature and pressure. Of this gas, about one-third is used by the ovens, about one-eighth by the blowing engines (if driven by gas engines), and about 10 per cent. is lost or used up in miscellaneous ways; thus about 45 per cent. of gas is available as surplus, or approximately 72,000 cu.ft.

Taking an average heat value of 100 B.t.u. per cu.ft., the horsepower developed by large gas engines would amount to about 30 brake horsepower for every ton smelted in 24 hr. If ordinary steam-blowing engines were already installed, the available surplus would drop to about 25 brake horsepower per ton of pig in 24 hr., and might even fall below this value.

With coke ovens the production of gas naturally varies with the quality of the coal, but an average figure is 10,000 cu.ft. of gas per ton of coal. The surplus gas, when regenerative ovens are employed, amounts to about 5000 cu.ft. per ton of coal. Where nonregenerative ovens are employed, the surplus is very much less, sometimes amounting to 2500 cu.ft. In these cases the high temperature of the escaping gases enables a good deal of steam to be evaporated by suitable boilers heated by the waste gases. Roughly speaking, about 1 to 1¼ lb. of steam will be generated for each pound of coal coked, and this steam may be used for driving steam turbines or other classes of engines. It will thus be seen how much power can be obtained by utilizing what was at one time a mere waste product.

*Extracts from the presidental address by W. C. Mountain delivered at the annual meeting of the British Association of Mining Electrical Engineers, in Shieffield, England, Sept. 27, 1912.

Someone has taken the trouble to count the number of pieces in an electric fan and to measure the length of wire used in the coils. An ordinary 16-in. fan is said to contain 1372 ft. of wire and to possess more than 750 pieces of material.—*Electrical World.*

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Ventilating System for Small Schools

By Charles A. Fuller

To design a mechanical ventilating system for a small school, which will come within the requirements of the state laws, often presents a more perplexing problem than the larger schools and auditoriums. With the small school first cost is usually of vital importance and must be kept down to the limit. The space allowed for apparatus is generally restricted, and operating costs must be studied carefully.

The accompanying plans are of a four-room school in which the ventilating requirements are well within the 30 cu.ft. of air per minute per pupil, prescribed by the state code, and the cost of the installation is only slightly more than of a direct-heating system.

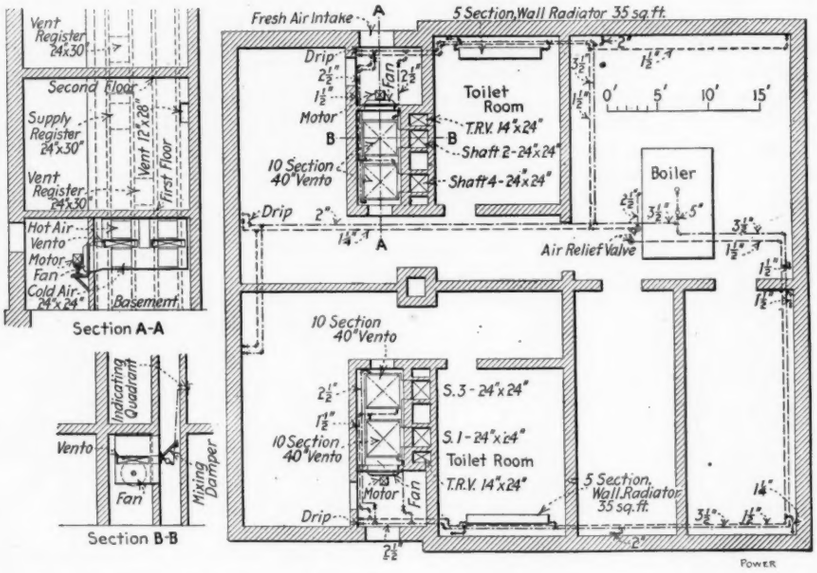


FIG. 1. PLAN OF BASEMENT, SHOWING BOILER CONNECTIONS AND INDIRECT STACKS

In each class room direct wall radiation compensates for the heat loss through the walls and windows, with no allowance for air change. Each of the four indirect stacks placed at the base of the warm-air flues is composed of 10 sections of 4-in. Vento cast-iron radiation, the sections being spaced on 4-in. centers, and each containing 10.75 sq.ft. of surface, or 107.5 sq.ft. for each stack. With the above spacing there is a free area of 0.35 sq.ft. per section and a total free area of 3.5 sq.ft.

Allowing 30 cu.ft. of air per minute per pupil and 50 pupils per room, gives 1500 cu.ft. per min. for each room. Dividing by the free area through the stack 3.5 sq.ft., gives a velocity of practically 430 ft. per min. The temperature rise at this velocity is 78 deg., that is, the incoming air will be heated from 0 to 78 deg.

Under these conditions the condensation will be 1.23 lb. per sq.ft. per hr. As the normal condensation in a direct radiator is approximately 0.3 lb. per sq.ft. per hr., 1 sq.ft. of indirect radiation in this case will be equivalent to

$$\frac{1.23}{0.3} = 4.1 \; sq.ft.$$

of direct radiation. Each stack is then equivalent to

$$4.1 \times 107.5 = 440.75 \; sq.ft.$$

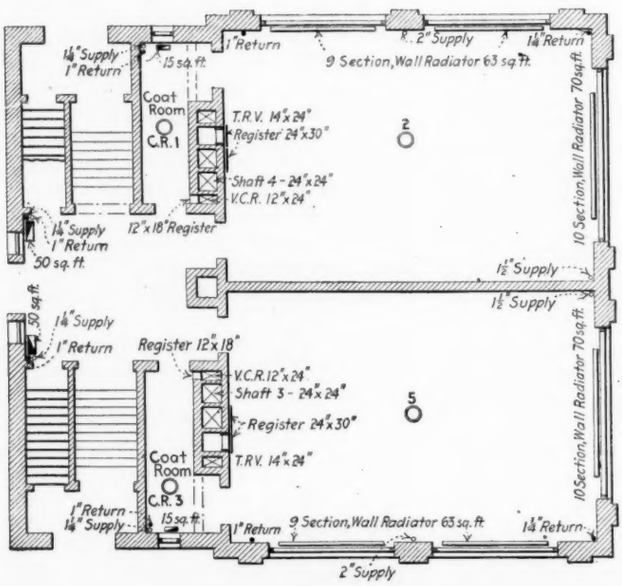


FIG. 2. FIRST FLOOR PLAN

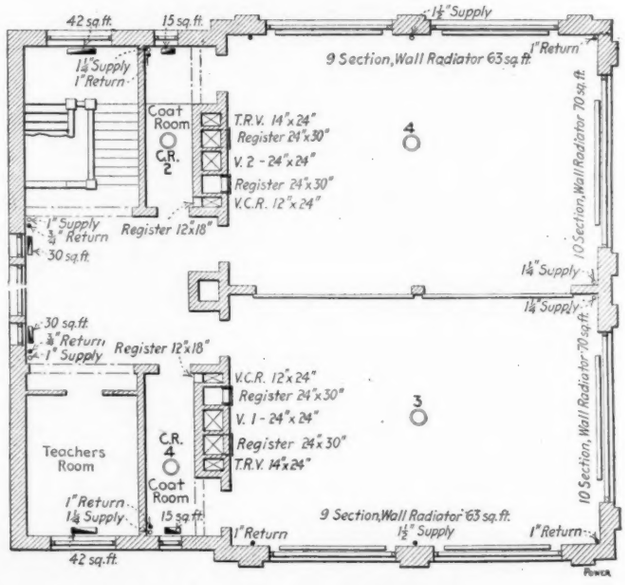


FIG. 3. PLAN OF SECOND FLOOR

of direct radiation and the four stacks 1763 sq.ft.

In the building there is 1158 sq.ft. of direct radiation and adding the 1763 sq.ft. for the stacks makes the equivalent of 2921 sq.ft. of direct radiation for the entire building. The boiler installed is of the cast-iron sectional type with a capacity to serve 4400 sq.ft. of radiation, an increase of 50 per cent. over the actual amount.

Two fans are installed, as shown in the basement plan. Each is of the disk type with a 36-in. wheel directly connected to a 1-hp. motor. When discharging into the atmosphere against no pressure, the fan at 600 r.p.m. is capable of delivering 10,000 cu.ft. of air per min. The resistance of the stacks and flues will cut this down by half, but even then the fan is amply large, as the maximum demand per fan is 3000 cu.ft. per min.

The horizontal ducts connecting the fans with the stacks and bypass openings could be omitted if the chamber into which the fan discharges were reasonably air-tight. This would lessen the friction of air flow and also decrease the first cost of installation slightly. The air would pass in the same manner either up through the stack to the vertical flues or through the bypass opening to the flue according to the position of the mixing damper. This damper is controlled by a chain and lever, with an indicating quadrant in the room into which the flue discharges. This is easily operated in the room to give any desired temperature of incoming air, regardless of outside temperature, if it is between 0 and 70 deg., without materially affecting the quantity of air delivered.

The supply registers are in the usual position, about 8 ft. from the floor, with the exhaust registers at the floor line. Separate exhaust registers and ducts are provided for the coat rooms and the toilet rooms in the basement. The vent flues are joined into two main flues passing up through the roof, and capped with 30-in. "Autoforce" ventilators. When toilet-room vents are carried up with the exhausts from class rooms, as in this case, care should be taken to insure that these tap into the main flue at a point considerably above the exhaust register connections from class rooms.

No air valves were installed on either the radiators or indirect stacks. All the air in the system is removed through two large automatic float air valves installed at a high point in the dry return near the boiler. The return end of each radiator and stack has an automatic impulse check valve.

The Victoria Falls & Transvaal Power Co., of South Africa, has almost completed its fourth electric power station on the Vaal River at Vereeniging. It will be one of the largest in that country.

Heating a School

By Fred L. Wagner

A few figures used in determining the size of the heating and ventilating plant in a school building may be of interest to members of the craft who are employed in such buildings. A building containing 30 classrooms will be considered. The first thing to estimate, is the probable heat losses which occur through the walls, windows and roof.

Heat Losses

Assume the building to be 190 ft. long, 80 ft. wide and 55 ft. high, containing 10 classrooms on each of three floors; the playrooms are in the basement. Each classroom will average about 200 sq.ft. of glass, or 6000 sq.ft. in all. The glass in the basement will approximate 1500 sq.ft., and the area in the halls, landings and entrance doors will approximate 1500 sq.ft., making a total of 9000 sq.ft. of glass surface.

According to the dimensions given the exposed wall and roof surface in this building would amount to

$$42{,}800 - 9000 = 33{,}800 \ sq.ft.$$

The heat loss through single glass is generally taken at 1.03 B.t.u. per sq.ft. per deg. difference in temperature between the outside and inside air, and that through ordinary brick walls at 0.2. In this case an outside temperature of zero was taken and an inside temperature of 70 deg. F., making a difference of 70 deg. Through the glass the loss per hour would be

$$9000 \times 1.03 \times 70 = 648{,}900 \ B.t.u.$$

while the loss through the walls and roof would be

$$33{,}800 \times 0.2 \times 70 = 473{,}200 \ B.t.u.$$

The total loss amounts to 1,122,100 B.t.u. per hr., which will have to be supplied by the heating plant to maintain a temperature of 70 deg.

Heat for Ventilation

The foregoing figures do not take into consideration the heat required for ventilation, which in schools is greater. In this school there are 48 pupils in a classroom, making a total of 1440, and 31 adults. An allowance of 30 cu.ft. of air per min. is required for each pupil and 50 cu.ft. per min. for each adult, making a total of 44,750 cu.ft. Increasing this amount one-fourth for hallways and basement gives 55,937, or in round numbers 56,000 cu.ft. per min.

Under ordinary pressures and temperatures 1 B.t.u. will heat 55 cu.ft. of air 1 deg. To find the number of heat units needed to heat the 56,000 cu.ft. of air from 0 to 70 deg., divide by 55 and multiply by 70;

$$\frac{56{,}000}{55} \times 70 = 71{,}272.7 \ B.t.u. \ per \ min.$$

or 4,276,362 B.t.u. per hr., which is greatly in excess of the actual heat loss in the building.

Surface of Coils

The amount of radiation needed in the blast coils will next be considered. Owing to losses of heat through ducts, walls and glass and infiltration of cold air, the air coming from the coils must be at a much higher temperature than 70 deg. In zero weather when all the coils are cut in, the temperature of the air leaving the coils often reaches 120 deg. and above, and in the present case 120 deg. was taken.

The amount of air needed for ventilation is 56,000 cu.ft. per min., and a fair average velocity for the air entering the heater is 1200 ft. per min. At zero temperature and with steam in the coils at 5 lb. pressure, it will take 20 rows of 1-in. pipe to heat the air to 120 deg.

Authorities on heating and ventilating agree that 1 sq.ft. of surface in blast coils with the air passing over it at the above stated velocity will heat 13 cu.ft. of air per min. from 0 to 120 deg. Using the figures at hand, 4307 sq.ft. of coil surface will be required. As school houses are heated only during the day, provision must be made for rapid reheating; therefore one-quarter will be added to the theoretical amount, which will increase the coil surface to 5384, or practically 5400 sq.ft. The number of sections and general dimensions of the heater may be determined from data given by the maker of the particular type chosen.

Size of Blowers

To obtain as nearly as possible an even distribution of air and heat, two fans are installed in a building of this size, the capacity of each fan being one-half of the quantity of air needed. There will be needed two fans, each having a capacity of 28,000 cu.ft. per min. To avoid noise the fans must be run at low speed, and to keep down the area of the ducts an air pressure of ¾ oz. would be used. This installation would require fans with wheels about 7x3½ ft. For driving each fan a 15-hp. engine or motor would be about right.

Boiler Capacity

The boilers installed to furnish the steam for the coils and auxiliaries should be of ample capacity to furnish the maximum amount of steam needed during the severest weather. In this case the boilers were rated on the supposition that they must supply 2 lb. of steam for every square foot of coil surface per hour. The 5400 sq.ft. of coil surface at the above allowance will condense 10,800 lb. of steam per hour. Dividing by 34½ lb., the evaporation per horsepower per hour from and at 212 deg., it will be found that the coils will absorb 314 hp.

per hr. during the coldest weather. The feed pump and vacuum pump will require approximately 10 hp., and the fan engines 30, making a total of 354 hp. in boiler capacity. Three return-tubular boilers of 115 hp. each will be of ample capacity to supply all the steam needed in this particular building. This boiler capacity will not only heat the air, but cover the losses through walls and windows.

[For a building of the dimensions given, the boiler capacity appears to be excessive. Rather than base the horsepower on the condensation in the coil surface it would have been better to rate the boilers on the heat required to supply the loss through walls and windows and to raise the air for ventilation from 0 to 70 deg. F. Adding the heat units required in each case, dividing by 1000 to obtain the pounds of steam and then dividing by 34½, will give the horsepower needed for normal operation.

$$\frac{1,122,100 + 4,276,362}{1000 \times 34.5} = 156.48 hp.$$

All exhaust from pumps and fan engines can be used in the heating system, but allowing a little extra capacity for rapid heating, 200 hp. of boilers should be sufficient.—EDITOR.]

Extravagant Steam Pipes

BY W. H. BOOTH

When an engine shows an initial pressure too far below boiler pressure, and its valves are properly set with a reasonable degree of lead, either the steam pipe is too small or the flow of steam is otherwise obstructed. It may be that a stop valve is small or that the engine ports open insufficiently. But in steam work, pipes are often too large or too small.

When a line supplies steam for heating only, and no power, it is a serious fault to make a pipe line so large that the steam pressure does not drop considerably at its extreme end, when perhaps only one or two radiators require supply. In a correct system of steam heating the condensed steam should gravitate back to the boiler room. Certainty of operation and flow of steam is guaranteed in a pumped return. There should be no need for traps. So long as steam can reach the most distant point of its use the pipe from the boiler need be no larger than will just supply this steam on the days of maximum demand.

In a complaint recently investigated where there was alleged excess of steam used, it was found that the steam main from a pair of boilers supplied no steam for power purposes. All boiler makers fit their boilers with steam outlet mounting blocks big enough to fit steam pipes which shall supply steam to an engine, within a reasonable distance, without loss of pressure of serious moment. But this does not demand that the steam main shall be of the size indicated if heating steam alone is wanted.

In this case the steam outlets on the boilers were 6 in. in diameter and the steam pipe was 6 in. for some distance, beyond which it branched off 4 in. each way. A great length of 4-in. main was laid underground in a brick culvert which was inaccessible. This had been full of water at sometime while the pipe had been carrying steam and the condition of the insulation may be guessed, for it could not be seen except at a few manholes, and there it was practically detached. The boiler pressure was 50 lb. and the first thing tried before making suggestions was to close all live outlets from the pipe and measure the pipe drainage all of which returned to one tank in the ground.

This test showed that the pipe condensation was the greater half of the mean consumption of feed water. The pipes were much too large, and it was advised that in place of a 4-in. main there should be a small wrought-iron pipe starting not over 2 in. and finishing at ½ in. It was quite sufficient that steam should reach the extreme end of the line at atmospheric pressure.

There were numerous similar wastes, duplicate steam pipes, a 7-in. main on another set of boilers where a 2-in. would have done the work of heating, if only a special 3-in. pipe to a noncondensing engine had been added. The exhaust from this engine would be then sufficient to have done much of the heating work. The engineer in charge knew that the conditions of the place were bad and that the steam mains were wasting half the coal, but his words were unheeded, and the writer's advice was regarded as too drastic to be followed. To cast out the big 4-in. and 7-in. mains, which had cost so much to put in was regarded as flinging away capital. All over the place water and steam were being run to waste and the management was weeping over the expense. They would spend thousands to show an economy of 5 per cent., but could see no sense in scrapping this pipe condenser half a mile long and saving 50 per cent. This is said to be a fair sample of the public institutions throughout England.

It appears to be usual to put in steam-heating mains as large as if they had a big engine to drive at their extreme end. For years one such institution ran all condensed water from pipes and heaters into the sewers and would have continued to do so, if not stopped by external pressure. Such places are built at an enormous public cost, excessive apparatus and pipes are put in and coal bills ever after form an excessive charge on the taxes. It seems hopeless to have such places put upon a sound basis. They advertise for good engineers, but when they get them, proceed to tie their hands.

Ventilating the Boiler Room

The old plant had outlived its usefulness and it was decided to put in new and larger boilers and uptodate machinery. The old chimney in the corner of the boiler room was not large enough and besides the space it occupied was badly needed for the new boilers. It was decided to build a new stack outside of the building wall. To prevent disturbance on the floors above only the lower part of the stack from the boiler-room floor to the ceiling was torn out. Holes, level with the floor above, were cut through both building walls and 12-in. I-beams resting on iron plates inserted. After putting in some cross-pieces to help support the stack, the lower portion was removed, leaving a splendid ventilating shaft.

As the plant is located in a wood-working factory, the boilers are inclosed by fire walls and naturally it would be a hot place in summer. As it is now, visitors are surprised to find at the front of the boilers the coolest spot in the plant.

There was some talk that the old chimney being used as a ventilator would affect the draft of the new stack, but this does not seem to be the case, and the draft from the new smoke-stack is all that could be expected.

R. CEDERBLOM.

Chicago, Ill.

Friction Loss in Wrought-Iron Pipe

There seems to be something wrong in Mr. Durand's chart on page 649 of the Oct. 29 number of POWER. A 10-in. pipe with a velocity of 1.5 ft. per sec. has a friction head of 0, and with a velocity of 1 ft. per sec., the friction head, according to the chart, would be negative.

WILLIAM KENT.

Montclair, N. J.

[Below the critical velocity in a pipe, the usual formulas for determining the friction head do not apply. The friction is almost negligible and to show it with any degree of accuracy would require a chart on a much larger scale than that prepared by Mr. Durand. According to the average values given in the table from which the curves were plotted, the friction head per 100 ft. of 10-in. pipe, with a velocity of 1 ft. per sec., is 0.043 ft. It is evident that the friction-head values do not hold to the same straight line below the critical velocity. In the chart under discussion this was neglected as only the extremities of the lines would be affected, and mention was made in the text of the inaccuracy of the friction values for a velocity of 1 ft. per sec. To avoid any misunderstanding, however, as in the present case, it would have been better to cut off the chart horizontally at 0.05 ft. friction head.—EDITOR.]

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POWER

Wanted: More State Boards

Only two states, Massachusetts and Ohio, seem to have taken up the subject of state control over boiler construction and inspection in genuine earnest. In both of these states the activity of the state boards has resulted not only in great benefit to public safety, but in awakening the engineering public to a proper sense of responsibility that rests with the man in charge of a steam plant.

Complaint is sometimes to be heard of unnecessary stringency in the examination of applicants for different grades of operating engineers. To test a man's intelligence and capacity it is necessary to make inquiries beyond the commonplaces of his daily routine. How else can one demonstrate his ability to cope with new questions? How else can value be placed on a man's intelligence and enterprise if not by inquiring into subjects that are more than ordinary?

The examining boards are doing a great service not only in protecting life and property, but also in uplifting the standard of deserving engineers.

To the Oiler

Some engineers have stepped directly from the boiler room into the engine room and shouldered the responsibilities of engineers; others have risen directly from oiler to engineer and assumed like responsibilities. But the man who has done the work of fireman and oiler before becoming an engineer is by far the more fortunate.

Every engineer should have had ample experience in firing boilers before he assumes charge of the plant equipment. It is indeed hard to make the average oiler who feels thoroughly competent to perform the duties of engineer realize this. He knows that firing is hard and dirty work; perhaps, too, he has been unfortunate enough to look upon the fireman as one of a lower order than himself, and so he aims to avoid the fires for fear he will lose a little of the dignity that goes with the job of oiler.

We would say to the young oiler who aspires to become a real engineer: Forget your dignity before some wise old flannel-shirted fireman pulls it out of you in a fashion that will make you feel cheap. Instead of holding aloof from the fireman, cultivate his friendship; he can tell you helpful things that are exceedingly hard to get through other channels of information. When you get on the night watch and have the bulk of your work done, do not select a dark corner to have your quiet smoke, but go clean a few fires; do just what you would do if it was up to you to run the place for the rest of the night, but be sure you do what is right. Some time when the load is heavy, if you can spare the time, clean a fire or two and find out what it is to properly clean them without "losing the steam." You cannot learn this from a book or by simply looking on.

When an engine is to be overhauled, a pump plunger packed, or when the cable from the damper regulator to the main damper breaks, leaving the damper shut during a heavy load, when such things—and they are many—happen, help to make the repair, get into the work "up to your neck" as the saying goes, for some day these things will happen in *your* plant. You will be held responsible for a shutdown and it is then that the experience gained at the expense of sweat and blistered hands will "pull you out of a hole" that the mere onlooker would be buried in.

When new apparatus comes into the plant, learn what it is, what it does and how it does it. Be "on the job" when it is first started and find out what its good and bad features are; new machines are usually "cranky" and it is well to get acquainted with their peculiarities—catalogs will not tell you.

To carry out these suggestions you must show a willingness to be of help and an eagerness to learn, and must do both tactfully or you will be offensive; hence you acquire something besides engineering skill.

There is no real short cut to becoming a good engineer, but the foregoing combined with technical study is the surest and quickest way.

Ambition First

At some time or other to every engineer comes the feeling that his ability is confined and he is filled with an impulsive desire to advance to a broader field of labor where his attainments can be more freely exercised and appreciated. This feeling may be induced by a desire for greater wealth; or, the gaining of added knowledge; or, perhaps, the combination of these for the greater power of accomplishment. But, whatever the inspiration, if the opportunity is presented and grasped at this time the realization of the desire depends on the test of the justification of self-confidence and the rapidity of progress. If, however, the opportunity be not recognized, or delay of indecision allows it to pass, the great majority readily fall back into the old ruts, the old habits of mental lethargy again dominate and the glorious vision of possibility is, perhaps forever, obscured by the ascendency of inherent laziness and lack of desire.

The condition of those who look back and realize "what might have been" had they not succumbed to the apathy of "lost opportunities" is pitiful. Constantly do they bemoan the hard luck which they consider has forced them to remain in their narrow environments, and seldom, if ever, do they admit that the advancement of their more favored brother engineer is the result of resolute, untiring effort in maintaining a fixed purpose.

It seems strange that anyone engaged in the stimulating, progressive vocation of steam engineering should ever be content to relinquish the effort of self-advancement, and the fact that so many do simply drift along can only be ascribed to hereditary habits of mental inaction so long indulged in as to easily become a potent force unless strenuously resisted.

The spectacle of the man bewailing the "lack of opportunity" would not be so pitiful were it not possible that it could be changed. The conditions demanded are simple. The methods to be used are easily explaind. There is only one factor that is repugnant, and that is the effort which, of necessity, must be made by the individual himself.

The great secret of advancement is ambition. Literally defined, it means the cultivation of "the desire of power, honor, fame and excellence." To cultivate the desire is simply to learn to concentrate one's energies on a single thing at one time. To some this is comparatively easy, almost natural, and their progress is rapid. To others, the development of the power of concentration requires an effort, and the reward will be directly proportionate to the effort expended.

Ambition does not distinguish between men or the quality of their social status. The uneducated laborer in the boiler room and the college-bred son of the owner are, insofar as the reward of properly concentrated effort is concerned, on an equality; the college education being merely a preparation for the concentration of future effort, while the field for bettering the comparative conditions is an incentive all in favor of the laborer.

Going to Extremes

POWER has published considerable relating to keeping power-plant records, and has tried to impress upon engineers the importance of knowing what the plant is doing. While the desirability of keeping plant records has not diminished, there is danger in overdoing the matter by incorporating items in the record sheet about which the chief engineer knows nothing or by not carrying the record far enough.

For instance, in one plant the kilowatt readings were taken once an hour and a load curve plotted from the readings. The highest peak shown by the load line was 1575 kilowatts, which was between 4:30 and 6:30 p.m. In reality the highest peak load was 2325 kilowatts, as shown on the switchboard instruments between the hourly reading.

It goes without saying that reading the switchboard instruments once an hour is not enough to enable the engineer or general manager to obtain accurate information as to what the generating units are doing. Fifteen-minute intervals should be the limit for reading switchboard instruments and this has been adopted by most large stations.

It is common to incorporate data items in a report sheet which have no bearing on the actual operation of the plant. For instance, one report sheet of a steam power plant contained the item "Average kw. per b.hp.," but there was no means of indicating the brake horsepower and the only way of obtaining the desired result was to use the builder's original test figure as a basis from which to make computations. If the steam end of the machine became inefficient the same brake horsepower would be used, although more steam would be used in developing the normal machine load.

An instance of making unnecessary labor for the engineer is an item reading "Tons (grs.) Coal Used," when all coal was purchased in short tons. Why it was desired to change from short to long tons in making out the record sheet is not apparent.

Items such as clerical labor, although there was no clerk; barometer reading, with no barometer; CO_2 reading, without a gas-analyzing apparatus, were also in evidence. Total pounds of water evaporated and water evaporated per pound of coal were also to be tabulated, but there was no means of ascertaining what the evaporation was other than a water meter that might at some time have been correct.

The aim should be to incorporate in power-plant record sheets only such data as bear on the plant operation. Anything else makes the report more complex and adds to the work of the engineer.

The N. A. S. E. is getting along in years. No. 1, of Boston, one of the earlier of the subordinate associations, celebrated its thirtieth anniversary on Thanksgiving Eve. Much has happened in power-plant engineering since the organization of this association. Who can predict what will happen between now and the completion of the first half century of its existence?

Speaking of a steel for boiler plate that, being of only moderate high tensile strength can be made remarkably ductile, an authority on the subject humorously said: "Why, a boiler made of such steel couldn't explode! It would simply bulge out and push you away from it."

Did you ever stop those leaks in that boiler setting? Better do so now; they are costing you unnecessary coal and labor.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Troubles of Quarry Pumps

Probably I have had my share of pump troubles, and a relation of them will no doubt prove of interest.

The equipment at the Nesbit quarry plant consisted of one compound duplex, 12x8-in. pump, one 6-in. duplex pump, one 8-in. centrifugal pump, submerged type and an 8-in. pulsometer. Except the pulsometer these were worked to their limit day and night and a breakdown meant a flooded quarry if their operation was delayed too long.

One night in January about 12 p.m. the superintendent routed me out of bed and stated that the night man had called him up by phone, saying the compound pump had stopped and that he could not start it and wanted me to hurry down to "get things going." Arriving at the pump house I found the water within a foot of the pump cylinders, and as there was not steam enough to start the pulsometer, I made a hasty examination of the pump and found the trouble to be in one of the water cylinders. Taking off the head showed that the nut and lock nut had backed off and the follower plate had turned sidewise in the cylinder; and as the piston rod came back on the return stroke it bent in a half circle, cramping the piston and stopping the pump.

Well it was up to me to get this rod straightened, so getting some oily waste, I wound it on the rod, and with a gasoline torch to help out, heated it until I could bend it back into place, using a track jack and some wooden blocks to bend it. As soon as the piston came loose and worked freely I repacked it, and after screwing on the nuts riveted the shaft slightly over the locknut to hold it until a better job could be done.

The water in the meantime had risen until it flowed into the cylinder and as it was nearly zero outside one can imagine how comfortable I felt. A day or two afterward I took out this rod and straightened it properly, drilling a hole through the locknut for a cotter pin. This held for several months when a new rod was secured.

Not long after this accident a heavy rain on top of a deep snow caused a bad flood and during the night the compound pump "threw" a valve. After shutting down the pump to replace the valve, the night man opened the high-pressure cylinder drain cocks and before he could get the pump started the water chased him out of the pump house. Another hurry call was sent for me, but by the time I arrived the water was over the pumps and two other pumps had stopped, the steam condensing too fast in the cylinders to give the required working pressure.

This had me "stumped" and I stood around not knowing what to do. Sometime later as I was standing in the engine room I happened to glance at the air compressor and it occurred to me that air would not condense like steam. Why not turn air into those pumps? Acting at once with the help of the superintendent and a laborer, we made a connection between the main air line and the steam main.

When the connections were completed, air was turned on, the two pumps that had stopped cleared themselves of water and started easily. Adding a little steam to the air kept it from freezing and the pumps were soon running at full speed. But the compound pump refused to start because the air went out of the open cylinder drain cocks too fast to allow the necessary pressure to accumulate in the cylinders. Not caring to do any fancy diving in 8 ft. of ice water, I "fished" for two hours with a bent rod before I closed three of the valves and got the pump started. The fourth valve had lost its stem and bonnet which had unscrewed and fallen off. It took three days to get the water down as the compressor could not furnish enough air to keep the pumps running up to their capacity, but as soon as the steam chests were clear of water I changed over to steam and soon had the water down to the working level.

Several times since I have had to use air but I have never delayed so long before making the change.

Another source of trouble was the breaking off of valve studs, level with the seats, in places that could not be reached to drill them out without dismantling the pump. The seats refusing to come loose unless cut out and extra seats were not always on hand. This I overcame by breaking out an arm of the seat so as to insert a double U-bolt that came up on each side of the center passing through two extra holes cut in a valve and projecting high enough to receive a spring and plate under the nuts. This allowed the valve to play as usual with very little loss of water and gave good service until they could be correctly repaired.

One night a rather peculiar breakdown happened to the centrifugal pump. This pump was driven by a 2-in. vertical shaft 28 ft. long with gears at the upper end and held up by a thrust bearing that was water cooled. The speed was 800 r.p.m. and the cooling water came through a ¼-in. pipe tapped from the discharge line. This ¼-in. pipe split near the bearing one cold night and a fine spray of water struck the shaft, forming an ice collar. The night man paid no attention to it, thinking it harmless as it only extended over about a foot of the shaft and seemed light like frost, but shortly before morning the shaft snapped in the center of the ice ring with a clean break that appeared as though the shaft had been cut with a hacksaw. This caused the engine to break its governor belt and by the time the night man got to it, it was doing its best to get out of the engine house.

The engine needed considerable repairing before being used again. Nothing broke, but things got twisted and sprung and the bevel gears lost all their teeth.

The company has since installed larger and better pumps and these troubles are a thing of the past as far as night work is concerned.

C. L. MALLERY.

Myerstown, Penn.

Cause of a Noisy Heater

An open cast-iron heater of standard make had been in use several years. At times lately this heater would emit a loud noise, accompanied by a heavy jar and violent fluctuation of the water level; this disturbance did not occur at regular intervals, but was most frequent when the boilers were using the most water.

The heater was connected to the exhaust of a cross-compound Corliss engine, and was located higher than the connection from the low-pressure cylinder. A valve in the exhaust line between the heater and engine was adjusted so as not to allow any more steam to go to the heater than could be condensed by the maximum amount of water admitted to the heater, the balance of the exhaust was used for drying. It was thought probable that the trap on the exhaust line did not operate, allowing water to accumulate in the line to the heater, then blow through causing the noise, but after examination this did not prove to be the cause.

When watching the exhaust head, located some feet above the heater, it was observed that at each occurrence of the noise the steam that was discharging

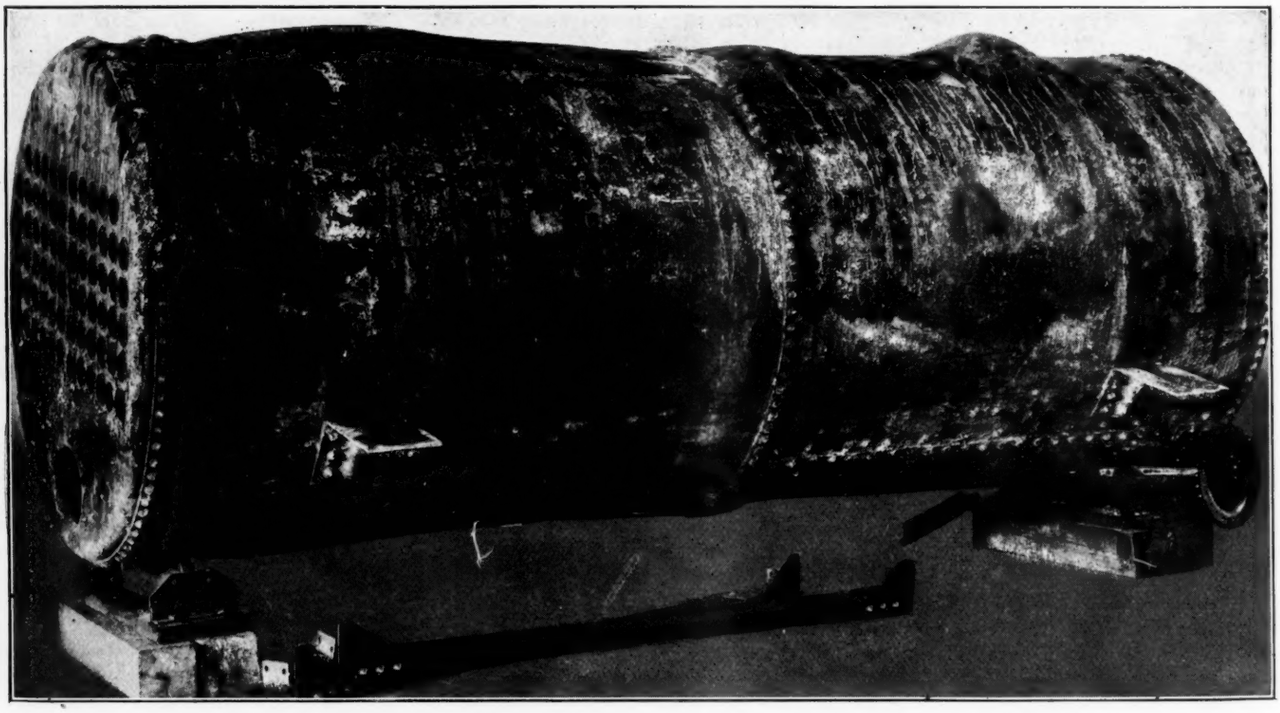
from the head would rush back down the pipe, indicating that there was a vacuum in the heater. This gave a clew that led to the close inspection of the balanced float-controlled valve on the water feed pipe. The stem of this valve was loose and corroded so as to take more than the usual power to make it move. When the valve closed it did not open again until the water level was almost below the float, when the latter was at its lowest position, the float would then open the valve which, being loose on the stem, would open wide, throwing a heavy spray into the top of the heater, thus condensing the steam and forming a partial vacuum that caused the noise and jar. After the valve was put in good order, we were not troubled with the noisy heater.

R. E. ROCKETT.

New York City.

Sediment Causes Serious Bags in Boiler

The illustration shows a 72-in. by 18-ft. boiler of our make (Houston, Stanwood & Gamble Co.) that was badly damaged in the plant of a large manufacturing establishment a short time ago.



SHOWING BAGS DUE TO SEDIMENT. BOILER IS UPSIDE DOWN TO REST ON TIMBERS

This boiler is butt-strapped, triple-riveted, made for 125 lb. working pressure and at the time of the accident had been in use only about a year. It is one of three boilers, set separately, which were installed at the same time. The factory is in a small town, and is probably the largest user of water in that town. Lately, a large amount of new water pipe had been laid, and these pipes had been dipped in coal tar. No adequate provision was made for preventing the débris that got into the pipes before they were laid, along with the coal tar, from being pumped into the boiler. Much water was used for other purposes and the strong currents thus created in the pipes allowed the deposits to be carried in large quantities to the boiler. The sediment in the largest bag was about 3 in. thick. The sediment is gray and powdery, feeling very smooth when rubbed with the thumb and fingers.

C. R. HOUSTON.

Cincinnati, Ohio.

Steam Boiler Legislation Needed

A 72-in. return-tubular boiler was supposed to have been cleaned at the end of the previous season. Whether the boiler was inspected or even looked into before being put into commission this spring, is not known. The fireman, however, could not blow it down with 10 or 15 lb. of steam and the engineer told him that it would be all right when the steam pressure got higher.

A new man took charge about the middle of the season and a stream about the size of a lead pencil was the best he could get through the blowoff. He was alarmed and wanted to investigate, but was not allowed to do so. The blowoff was not protected; and sand and scale about 5 in. deep rendered the shell-heating surface practically valueless. Apparently the only thing that prevented a disastrous explosion was that the boiler could make all the steam necessary with very moderate firing.

Doubtless many other similar conditions exist in localities not protected by law, but a few such exposures should bring about some necessary steam-boiler legislation.

A. J. CARR.

Portland, Maine.

Cotton Ropes for Power Transmission

The first rope drives were in Dundee, as that town is the center of the hemp and jute trade. The ropes employed were hemp and even stiffer than manila hemp for they were more closely spun or twisted.

As a textile district, Lancashire was more important and much busier than Dundee, and rope driving became very popular. But Lancashire, being built on cotton, would have nothing to do with hemp, but must employ cotton ropes. Just prior to this rope-driving enthusiasm the rise of numerous engineering firms had taken place.

Their proprietors were men of little technical knowledge and the wheel gears in the works (i.e., toothed gears) were as rough and as noisy as themselves. These undesirable gears accelerated the introduction of the American driving system with large wide belts on big driving pulleys and some excellent belt drives were installed.

Rope driving superseded belts almost entirely in Lancashire, but they endured

longer in Yorkshire, for that was the home of the woolen industry and the Yorkshiremen preferred to do anything rather than follow the lead of Lancashire. The wide leather belts did not run badly, but had an objectionable flapping, air-clapping trait even when running up hill onto the driven pulley; this is unknown with ropes.

Even within recent years, although ropes have long been almost the sole system of driving, badly designed rope drives have been installed by electrical engineers whose ideas of driving were at variance with the characteristics of electric machines. Without hesitation they would set a dynamo low down on the floor and drive it from the large rope pulley, the slack side of the ropes running down to the driven pulley. Whenever a rope runs down hill in that way it tends to throw itself into a loop instead of hanging down in a catenary curve and enveloping more, instead of less, than half the circumference of the driven pulley. Matter in motion tends to proceed in a straight line, and as the rope comes over the top of the driving pulley it will try to keep a horizontal path, but it is hauled down by gravity and by the preceding part of the rope. If the pulleys are 30 ft. apart and the rope is traveling at 6000 ft. per min. it will cross the 30-ft. gap in 0.3 sec. But in 0.3 sec. a body will fall

$$16 \times 0.3^2 = 1.44 \text{ ft.}$$

The vertical distance between the top of the, say, 30-ft. driving drum and the 6-ft. driven drum is perhaps 12 or 10 ft. and so there is an effort for the rope to make a loop or hump of 8 or 10 ft. above the driven pulley. Is it surprising that ropes will run around a small driven pulley without even touching it? In a traction load these rope humps are especially bad if the generator has no flywheel effect. But with a well designed system the results are very good. One of the best of nine traction stations under my charge, so far as freedom from breakdowns and other troubles is concerned is the one rope-driven station.

In Lancashire practice there may be 40 ropes on the large pulley which are led off to the various rooms of the mill, so many ropes to each driven pulley. The ropes to the highest pulley go up at a very steep angle, those to the lowest may run slightly down hill. If the ropes be looked at when running, only the lowest set will ordinarily have any swaying motion; they run toward the driven pulley with a slight waving motion. But all the other ropes hang steady, and have a slightly reversed curve onto the faces of the driven pulleys.

Opinion is divided as to whether a rope should have three or four strands. These stranded ropes cannot pull out of shape for the third strand has nowhere to go except the place intended for it. Three strands of a four-strand rope can throw off the fourth weakling and put a rope out of shape.

Personally I could never see any good in making a four-strand rope around a straight centered core that must somewhat easily break when the twisted part of the rope stretches. But I have used four-strand small ropes, or rim bands on mules and they acted reasonably well.

It is usual to dress cotton rope slightly with plumbago and black lead. This serves to hold down fluffiness.

In Yorkshire, square ropes of leather have been used. They are made from 1¼- or 1½-in. strips of leather formed into a square and are easily pieced, using French nails to hold the strips together. They do not lie even but cling to both sides of the grooves and present all sides upward at different times for they do not always keep each face square but twist in themselves. They drive well and are very satisfactory.

A few leather-covered wire ropes are used. These have a ⅝-in. steel-rope core to do the work and an external cover of 1½-in. leather washers which may be threaded on before splicing, the remainder being put on by cutting an acute diagonal through them and slipping them on. Such washers may be punched out of any old belting. These ropes do well and run for years. I know of cotton ropes, too, which have run for 25 years without a change.

A speed of 4800 ft. per min. is regarded by some as the maximum best speed. Beyond this velocity it is held that centrifugal force, increasing as the square of the velocity, adds more to the stress in the rope than the speed increases the capacity. At usual working tensions these two factors cross one another on the performance curves. Others, including one of the leading makers of cotton ropes, say that centrifugal stress does not exist in practice, but that the rope is gripped in the rim grooves sufficiently to hold it there and it suffers no tension from centrifugal stress, but that action helps the rope to leave its groove when the time comes for it to do so. And by this reasoning speeds of 7000 ft. per min. are held to be safe. But there must be some velocity at which the rope will not grip in its groove all the way around the wheels, and a still higher velocity where it will not even touch the grooves. These are for the mathematician to work out. It appears certain that at 7000 ft. the danger point has not been reached.

Manila, in this country is not favored as a driving rope. It is claimed that it is rough fiber and will grind itself away quickly. But I well remember that the late C. W. Hunt would not agree to this. He contended that a manila rope properly laid up of selected fiber and lubricated internally was superior to cotton. Manila is a stronger fiber, but cotton is very much more elastic and pliable. Every one of its many threads is of identical size and in the system of laying the threads parallel and untwisted in little strands there is obtained the maximum of strength. An ordinary rope will be rated at 50 hp. at 5000 ft. per min. This implies a working stress of only 350 lb. in a rope apart from the slack-side tension and the centrifugal tension. Ropes are run with a wide margin of strength and this is considered to pay in the long life it gives to the ropes.

Ropes run best when they always expose the same part uppermost, for they bed well to their grooves. But occasional ropes in a set will rotate about their longitudinal axis and they are thus constantly kneaded between the sloping faces of the grooves. They then become fluffy and short lived. Nobody seems able to discover why they rotate, though it is possibly due to the failure, when splicing, to lay them out and take off all superfluous twists tending to form kinks. New ropes should be hung with heavy weights to stretch them before splicing and a splice should be very long, at least 72 rope diameters in length. A well spliced rope joint cannot be distinguished after a short run.

Rope driving is being displaced in some cases by electric driving and greater production is claimed, but no very startling evidence has been forthcoming. The electric drive is usually compared with some old drive of 30 or 40 years' standing. Many present electric drives are little else than the use of a motor on the mill shafts instead of the rope pulley and no shafting is abolished, or where shafts are abolished any stock motor has been dumped down on the floor and set to drive a machine through a short stiff belt. These crude methods have done harm to the electric drive which should be made to do things that cannot so well be worked out mechanically.

As regards driving efficiency it was customary to say that where good gearing would drive a given mill with 1000 i.hp. a belt would demand 1050 i.hp. and ropes 1100 i.hp. But the ropes went with quicker running engines and higher pressures and they introduced of necessity such large rim wheels that the flywheel effect was many times that of the old spur rim, and the machinery ran without perceptable variation of hum, and less waste and bad work resulted. And ropes did not cause heavy breakdowns. If a rope fails it fails gradually, stretches abnormally, relieves itself of its share of work, and if badly neglected it throws out short wisps and loose tails of yarn which strike a safety bar and with a mere touch will stop the engine before the rope can run amuck.

W. H. Booth.

London, S. E., England.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Nonreturn Stop Valves

The editorial on this subject in the Nov. 5 issue was indeed interesting. I agree with everything said, because my experience with nonreturn valves corroborates the editorial.

In our plant we generate nearly 25,000 boiler horsepower in four boiler houses, which are all connected by a common steam main. If these boilers were not protected by nonreturn valves, and a tube should blow out, one can easily imagine the loss in time which would result. It might not only shut down the boiler house where the accident occurred, but even one of the other houses, if the valve in the line was not quickly closed. These conditions were considered sufficient reason for installing nonreturn stop valves on every boiler in the plant. This was done four years ago.

Besides the possible saving in time, there is, as the editorial pointed out, an additional reason for their use in that they are insurance against accident to workmen and hence are of value to the company. If a fireman should be burned, he might, after suffering pain and loss of time, recover damages from his employer. While the use of nonreturn valves will not keep tubes from blowing out, they minimize the danger attending such accidents. Consequently, they would undoubtedly cut down the number of accidents, and the number of resulting suits for damages, thus proving a saving in another way besides that of time.

An experience of mine illustrates this. A boiler was out of service for cleaning. In the drum, the manhole of which faced the adjacent boiler, was a boiler washer, and an inspector was working in the furnace. The boiler next to it blew out a tube and, although the men were badly frightened, they suffered no injury. If this boiler had not been equipped with a nonreturn valve, which operated instantly, these two men would undoubtedly have been boiled to death before the stop valve could have been closed.

These valves are reliable, too, for we have had five tubes blow out since they were installed, and each time the valve closed instantly. Again, when firing up a boiler there is nothing to worry about when it comes to cutting the boiler in on the line. When the boiler pressure equals the pressure in the header, the valve opens automatically.

Aside from all the obvious advantages of the nonreturn valve that mean a probable saving in time and money, and which certainly justify the expense as insurance there is a more important consideration, that is, the humanitarian side. Wherever it is possible to safeguard workers by installing safety devices it should be done. The automatic nonreturn stop valve comes in this category, and so its use along with other safety devices, should be required by law.

J. F. MOWAT.

Joliet, Ill.

Mr. Hawkins' letter and the editorial on the above subject is timely and should receive the approval and support of every engineer. I have had as many cases to deal with as Mr. Hawkins mentioned and a shutdown was the result in each of them..

The most serious accident was to a large tandem-compound engine in a power plant containing five other engines rated from 150 hp. to 1000 hp. running both direct- and alternating-current machines for street-car service and lighting.

The trouble started with a hot crosshead and before the guide could be loosened and the engine stalled, which was serious as there were no spare machines to carry the load. The crosshead parted at the wristpin and split the cylinders from end to end, broke the yoke connection between the two cylinders, "punched" out the cylinder head from the high-pressure cylinder, broke the end of the low-pressure steam chest and, worst of all, broke off the flange from the upper end of the throttle valve. The only means left to stop the flow of steam that followed the breaking of the flange was to close the stop valve on the engine steam pipe, located near the steam main, some 20 ft. from the floor. The break in this 6-in. pipe quickly filled the room with steam.

The oilers and assistants closed the throttles on the other five engines. One man had to crawl on his hands and knees to reach the engine-room door, as the windows were all covered with heavy wire screens. This shows how quickly a large room can fill with steam.

There were seven 72-in. by 18-ft. boilers in operation carrying a steam pressure of 115 lb. All the safety valves were fitted with a 3-ft. piece of pipe pointing toward the roof, which was about 12 ft. from the top of the boilers. Five of the boilers were connected at the front nozzle to an equalizing main from which steam was taken to run the condenser's auxiliaries and feed pumps. The stop valves on the risers were never packed up to the time of this accident since the plant started five years before. The wheels of these valves could hardly be turned with a 24-in. wrench, although the stems were leaking. The boiler room was already filled with steam above the boilers. But the flow of steam had to be stopped and the assistants, oilers and everybody in the plant except the firemen were working hard to close the valves. The valves were finally closed, but almost immediately the safety valves on two other boilers started to blow to add to the already uncomfortable heat of the room. Luckily the firemen had the fires banked just as the valves opened. The plant was idle for about three hours. Some machines were badly affected by moisture from the steam and were run some time without load to dry them out.

One year later the connecting-rod and crosshead on the high-pressure side of a 500-hp. direct-connected railway unit dissolved partnership by breaking the strap and the same experience was repeated. This was the third engine wreck in this station, but a nonreturn valve was unheard of at that time. Numerous other accidents have happened with some narrow escapes from injury and death.

This is only one plant out of thousands which are experiencing just such troubles, and while the public is suffering the effects of delays in service, the management busies itself by repairing and replacing the old valves only to meet another shutdown, when under a heavy overload some engine "takes a drink" from a foaming boiler and "punches" out the head or bursts a receiver and many other numerous accidents.

Many public-service corporations are installing nonreturn valves in their new plants and replacing the old valves with them as rapidly as repairs are necessary, but in some smaller plants where the danger to life and property is even greater a suggestion on the part of an engineer that nonreturn valves should be installed in place of a broken stop valve, is usually met with a retort that the valves have given satisfaction for 20 years and such accidents may never happen, and it is only by chance that such breaks do occur.

But these accidents are becoming too frequent of late years with the increase of boiler pressures, and like the old lap-seam boiler and the old style safety valves the stop valve must give way to be superceded by a nonreturn stop valve. Engine steam pipes should also be equipped with a nonreturn stop valve between the throttle and the steam main which would cut out an engine as soon as the cylinder loses a head.

It is the engineers' duty to put this fact up to the managements of their companies and if they refuse to heed the warning then the license laws should be amended and legislation sought, which would prevent a new plant being put into service unless such stop valves were provided and that no stop valve on any boiler now in service, when once removed for repair should be put in such service again but be replaced by a nonreturn stop valve instead.

I hope the state of Massachusetts will soon have such an amendment to its boiler laws. I wish to thank Mr. Hawkins and POWER for bringing this important subject to the engineers' attention, for there are many in the business who have not yet seen the necessity of using the nonreturn stop valve on boilers.

R. A. CULTRA.

Cambridge, Mass.

Diagram for Quickly Finding M. E. P. at Various Cutoffs

The table of mean forward pressures for various cutoffs and initial steam pressures, given by Wallace S. Fowler, on page 632 of the Oct. 29 issue, suggests that I might add a time saver of my own along the same line. It is incorporated in the accompanying diagram, which I plotted some years ago, because I found such a diagram to be of material aid in my work.

But little explanation is required, except to say it is based upon the regular mean pressure formula

$$P_m = P_1 \frac{1 + hyp.\ log.\ R}{R}$$

with which everyone is familiar.

As an example of the use of the diagram, let the initial steam pressure be 164.7 lb. abs., the back pressure 4 lb. abs., and the cutoff ⅓. Also let the diagram factor be 0 S, all as given in Mr. Fowler's example. From the curve we have the ratio of absolute mean pressure to absolute initial pressure, for ⅓ cutoff, given as 0.7. So the absolute mean pressure is 0.7 of the absolute initial, or

$$0.7 \times 164.7 = 115.29\ lb.$$

Deducting the same absolute back pressure of 4 lb., leaves a theoretical mean effective pressure of 111.29 lb., and multiplying by the assumed diagram factor of 0.8 gives 89 lb., agreeing with Mr. Fowler.

One of the greatest conveniences of such a curve is the possibility of working the problem backwards, to determine the necessary cutoff. The application of the "hyp. log. formula" is difficult for this.

Let us assume we must have 89 lb. m.e.p. with 164.7 lb. abs. initial and 4 lb. abs. back pressure. With a diagram factor of 0.8, as before; what must the cutoff be?

The theoretical mean effective pressure must be

$$89 \div 0.8 = 111.25\ lb.$$

With 4 lb. back pressure, the mean absolute forward pressure must be

$$111.25 + 4 = 115.25\ lb.$$

If the absolute initial pressure is 174.7 ib. the ratio of the absolute mean to the absolute initial pressure must be

$$\frac{115.25}{164.7} = 0.7$$

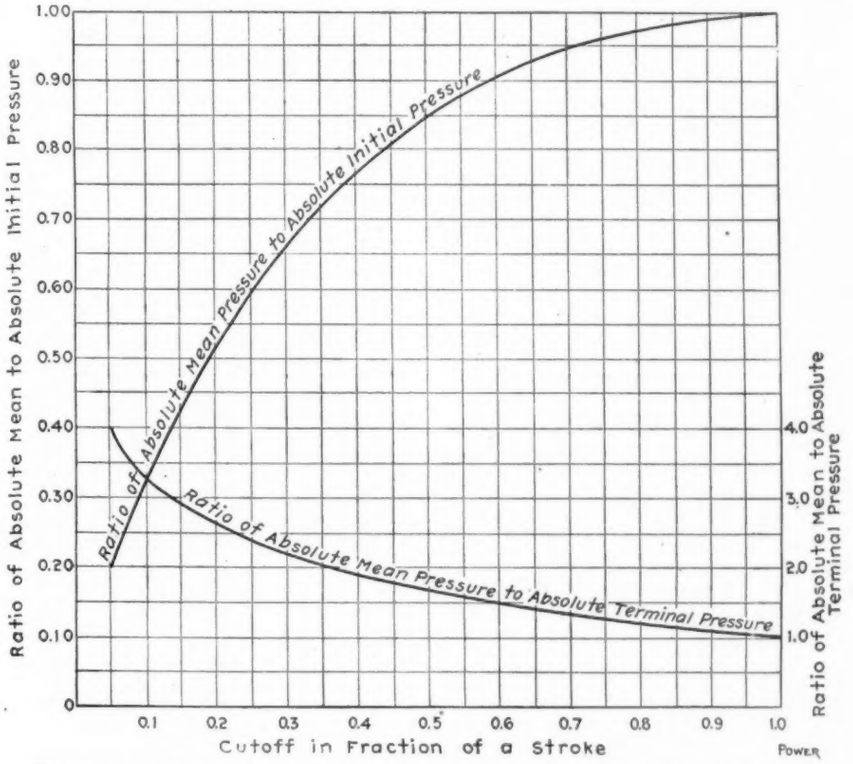


CHART FOR QUICKLY FINDING MEAN EFFECTIVE PRESSURE AT VARIOUS CUTOFFS

A glance at the curve shows that this ratio of absolute mean to absolute initial pressure corresponds to a cutoff of 0.333 or ⅓. This agrees with the original problem.

Further, let the problem be to determine the necessary initial pressure so that, with ⅓ cutoff and 4 lb. back pressure and 0.8 diagram factor, we may have 89 lb. m.e.p. As before, the theoretical mean effective pressure equals

$$89 \div 0.8 \times 111.25\ lb.$$

and the absolute mean forward pressure equals

$$111.25 + 4 = 115.25\ lb.$$

The curve shows that ⅓ cutoff gives the ratio of absolute initial to absolute mean pressure of 0.7, as before, so the absolute initial pressure must be

$$115.25 \div 0.7 = 164.7\ lb.\ abs.$$

as before. If this initial pressure is not satisfactory, it might be desirable to alter the cutoff, so one can glance at the curve and find that with, say ¼ cutoff, the ratio of the pressures is 0.596, so the initial pressure would then have to be

$$115.25 \div 0.596 = 193\ lb.\ abs.$$

Or, if a lower initial pressure is desired, a cutoff of say 0.4 can be selected, giving an initial pressure of

$$115.25 \div 0.766 = 150\ lb.\ abs.\ (about)$$

It was for just these "backward jugglings," to determine the best conditions that I originally plotted the curve, as a curve gives all the intermediate values, which is rather impracticable in a table. And, again, a curve gives one a clear mental picture of the various relations.

Another use for the curve is its application to compressed air. Isothermal air compression is the same as theoretical steam expansion if exhaust occurs at terminal pressure. For obtaining the efficiency of compression of an air compressor referred to the isothermal, it is necessary to compute the isothermal horsepower. Let us assume compression from atmosphere to 100 lb. gage. The ratio of compression is

$$\frac{114.7}{14.7} = 7.8$$

This corresponds to the ratio of expansion in a steam cylinder, so the cutoff is its reciprocal or

$$\frac{1}{7.8} = 0.1282$$

The curve gives the ratio of absolute mean to absolute initial pressure for this cutoff, as 0.391. As in air compression, the final, or discharge pressure corresponds to the "initial" pressure of a steam diagram, the mean pressure is

$$0.391 \times 114.7 = 44.9$$

Also, as in air compression the initial pressure corresponds to the back pressure of a steam diagram, we must deduct the initial pressure, which in this case is atmospheric. So 44.9 — 14.7 = 30.2 lb. m.e.p. for isothermal air compression from atmosphere to 100 lb. gage.

This last problem may also be worked from the lower curve of the diagram. A cutoff of 0.1282 corresponds to a ratio of absolute mean pressure to absolute terminal pressure, of 3.055, and as in air compression the initial pressure corresponds to the terminal pressure of a steam diagram, we have:

$$\textit{absolute mean pressure} = 3.055 \times 14.7 = 44.9 \textit{ lb.}$$

and subtracting the atmospheric pressure, as before, we have

$$44.9 - 14.7 = 30.2 \textit{ lb. m.e.p.}$$

It is difficult to read the curves as closely as this, but the diagram is very handy for approximate calculations, and if desired it may be replotted to a larger scale.

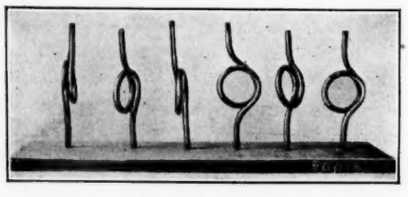
R. S. BAYARD.

New York City.

Rosin for Bending Pipe

C. R. Summer's letter in the Sept. 3 issue gave me an idea of how to make siphons for steam gages. He said he had never seen rosin tried for bending pipe but did not see why it would not work as well as for bending boiler tubes.

It does work well. I made six as perfect siphons, or goose necks as they are



SAMPLES OF BENDS MADE WITH ROSIN-FILLED PIPE

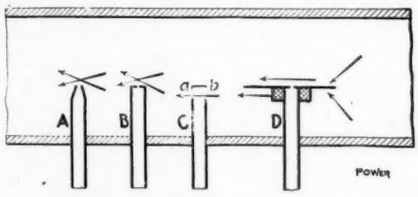
sometimes called, as any engineer would wish to have his steam gage fastened to, without a failure. I used ¼-in. black iron pipe and bent it cold over a 2½-in. pipe held in a pipe vise, threading both ends of the ¼-in. pipe before bending. I put a cap on the pipe before filling it with melted rosin, and before bending. The other end was capped to prevent spoiling the threads. A pipe about a foot long, which will go over a cap, will help in bending. A bunsen burner was used to melt the rosin and clean the finished siphons. The siphons were without a sign of a flat place; they are shown in the accompanying engraving.

H. S. SWEETSER.

Medford, Mass.

Pitot Tube Ends

On page 390, Sept. 10 issue, I noticed a discussion as to the merits of different forms of pitot tube ends, by G. G. Crewson. This is all very interesting to me as I have had some experience along the same lines. I tried the static tube ends as shown at *A, B, C* and *D* in the accompanying sketch and found that *A* and *B* would not give duplicate readings for conditions that were apparently constant, because of cross-currents, as indicated by



DIFFERENT FORMS OF TUBE ENDS

the arrows. The end *C* was very satisfactory but the least variation of the line of the hole *ab* from the line of air movement gave an appreciable error. However, the disk on the end of the tube, directs the air perpendicularly over the tube end and serves to eliminate the effect of cross-currents. The two last check each other well when *C* is carefully set. The ease with which *D* may be set (assuming, of course, that it is possible to get the disk in the duct at all) makes this modification very satisfactory.

J. D. HOFFMAN.

Lincoln, Neb.

Scale in Steam Turbines

The editorial in the Nov. 19 issue of POWER regarding scale on turbine blades furnishes an interesting topic for discussion. On first thought it would seem impossible for scale to form when the steam is properly superheated and a separator is installed in the line. Nevertheless, it seems to be a fact that this sometimes happens, and there must be some explanation.

There are only three ways that scale could get into a turbine under any circumstances, viz., through the steam inlet, through the journals and through the exhaust outlet. The latter can certainly be eliminated as a source, for there is a constant current of steam in the opposite direction. Leakage of scale-forming water through the journals is a possible cause of scale.

It is easily understood how scale could form on a turbine blade provided saturated steam were used which had not passed a separator or where the separator was of an inefficient type. Let us now consider what might happen in case superheat were carried.

As it leaves the boiler proper, even under normal conditions, steam will contain sometimes 3 to 5 per cent. of water. Bad, scale-forming water it probably is, too, due to concentration in the boiler. This water is in a finely divided state suspended in the steam. As the steam enters the superheater, each small particle of water it carries is evaporated, leaving a microscopic particle of scale not nearly heavy enough to drop out of the rapidly moving current. What happens then? These minute dust-like scale particles come in contact with everything which the steam hits and are for the most part carried out with the steam to the turbine.

What effect will a separator have on these particles? All common forms of separators are intended to separate liquids from gases. They are based on the principle that the momentum of a water or oil particle will carry it against the baffle while the steam current passes around the sides. They are not built to completely remove dust from a gas current.

We thus have these extremely minute scale particles passing by the separator to a large extent, and hitting, with the steam, against the turbine blades. Some of them stick and in time form a dense hard scale as described in the article to which reference was made. I believe that if the baffle-plate of the separator were inspected it would be found to be covered with the same dense hard scale which was found on the turbine blades. This would, of course, make it inefficient for removing water from saturated steam, as the ribs might be entirely plastered over. Ordinarily a separator is self-cleaning as the water removed from the steam is continually washing its surfaces.

Now, supposing the above explanation is the correct one, and it certainly seems plausible, what is the remedy? Naturally to put an efficient separator between the boiler and superheater, thus preventing any scale-forming water from leaving the boiler. Care should be taken to install one which is known to be efficient and the maker should be consulted as to the requirements. With such an arrangement, I believe the difficulty would be solved. It would at least be interesting to know what results are secured where such an arrangement exists.

EDWARD H. ROBIE.

Philadelphia, Penn.

On Thursday evening, Nov. 14, the Engineers Orchestra, of Worcester, comprised principally of members of Worcester No. 4 and H. E. Stone Association No. 32, N. A. S. E., gave its first public concert of the season. The program was well rendered and the concert was well attended.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Rotary Pumps

What is the advantage of rotary over centrifugal pumps used for private fire service?

R. S.

Rotary pumps can be operated under variable heads and speeds and are therefore more desirable as fire pumps in cases where the power is likely to be supplied at an uncertain speed.

Friction of Shafting

Will it take more power to drive 50 ft. of shafting from one end with a load on the other end than to drive the same shafting with the same load near the driven end of the shafting ?

H. P. A.

Yes; more power will be absorbed by the shafting when transmitting power from one end to the other than when a part of it runs idle, as the transmission of a load tends to throw the shafting out of line, thereby increasing the friction of the bearings.

Packing for Gasoline Joints

What is a good material for packing glass gage fixtures of oil or gasoline tanks?

J. H.

Ordinary rubber gage-glass packing rings, shellac varnished and dried before use, will be found to resist the solvent action of oil or gasoline; and when thus protected the rubber retains its elasticity and remains a very reliable packing material for a considerable length of time. Rubber rings used in this manner should be of proper size to fit snugly around the glass, but without being stretched so as to separate the shellac coating.

Cost of Coal vs. Wood as Fuel

What would have to be the price of average good bituminous coal per ton to equal the cost of dry wood at $4.25 per cord of 3800 lb., used as fuel for steam boilers?

H. D. B.

Cord wood that has been dried for about one year and containing 20 to 25 per cent. moisture, will have about 45 per cent. as much heating value per pound as bituminous coal of average quality. A cord of such wood weighing 3800 lb. therefore would have as much heating value as $3800 \times 0.45 = 1710$ lb. of coal, and the cost of it being $4.25 would be equivalent to coal costing

$$\$4.25 \times \frac{2000}{1710} = \$4.97 \textit{ per ton of } 2000 \textit{ lb.}$$

Electrical Transmission

What would be the number of brake horsepower required for driving a direct-current generator suitable for a delivery of 40 brake horsepower by electrical transmission 150 ft. from the generator?

H. L. H.

The average efficiencies of generators and motors of the sizes required would be about 90 per cent. for each. Assuming that transmission of current is by wiring of ample cross-section for the voltage, the efficiency of transmission would be about 97 per cent., and the power required for driving the generator would be about

$$\frac{40}{0.9 \times 0.9 \times 0.97} = 50.91 \textit{ b.hp.}$$

Size of Pump Air Chamber

What should be the size of the air chamber for a double-acting, single-cylinder, 6x10-in. boiler-feed pump?

R. E. B.

The content of the air chamber should be not less than three times the pump displacement for a single stroke. The area of a 6-in. diameter piston being

$$6 \times 6 \times 0.7854 = 28.274 \textit{ sq.in.}$$

the displacement per single stroke would be

$$28.274 \times 10 = 282.74 \textit{ cu.in.}$$

and the content of the air chamber should be

$$282.74 \times 3 = 848.22 \textit{ cu.in.}$$

If the air chamber is made cylindrical 8 in. in diameter with the upper end hemispherical, then as the volume of a sphere may be found by the formula

$$\textit{vol.} = \tfrac{1}{6} \times 3.1416 \times \textit{cube of the diameter}$$

for a hemisphere 8 in. dia.

$$\textit{vol.} = \frac{\frac{1}{6} \times 3.1416 \times (8 \times 8 \times 8)}{2} = 134.04 \textit{ cu.in.}$$

This would leave

$$848.22 - 134.04 = 714.18 \textit{ cu.in.}$$

to be contained by the cylindrical part of the air chamber. The diameter being 8 in., the cross-sectional area would be

$$8 \times 8 \times 0.7854 = 50.265 \textit{ sq.in.}$$

hence the length of the cylindrical part should be

$$\frac{714.18}{50.265} = 14.2 \textit{ in.}$$

Pumpage per Hour

A duplex pump having double-acting water pistons 10 in. in diameter and making a 12-in. stroke has piston rods 1¼ in. in diameter. What would be the net discharge in gallons per hour, making 52 double strokes per minute, allowing 5 per cent. slip ?

E. R.

The outer side of each piston would have an area of 78.54 sq.in. and the net area of the piston-rod side of each piston would be the same less the cross-sectional area of its 1¼ in. diameter piston rod, i.e.,

$$78.54 - 2.40 = 76.14 \textit{ sq.in.}$$

Therefore, the piston displacement for one double stroke would be

$$(78.54 + 76.14) \times 12 = 1856.16 \textit{ cu.in.}$$

or

$$\frac{1856\ 16}{231} = 8.035 \textit{ gal.}$$

per double stroke, which with 5 per cent. slip is reduced to

$$8.035 \times 0.95 = 7.633 \textit{ gal.}$$

and with the pump making 52 double strokes per minute, the net discharge would be

$$7.633 \times 52 \times 60 = 23{,}815 \textit{ gal. per hr.}$$

Dimensions for 15 Gal. Tank

What should be the length and thickness of material of a 15-gal. gasoline tank made of copper if 12 in. diameter and provided with one inward and one outward bumped head?

W. F. B.

One U. S. gallon is 231 cu.in. and 15 gal. would be

$$231 \times 15 = 3465 \textit{ cu.in.}$$

The cross-sectional area of 12 in. diameter would be

$$12 \times 12 \times 0.7854 = 113.0976 \textit{ sq.in.}$$

therefore 1 in. in length of the tank would contain practically 113.1 cu.in. and to contain 15 gal. or 3465 cu.in., the length of the tank would have to be

$$3465 \div 113.1 = 30.63 \textit{ in.}$$

or practically 30⅝ in. To be laid on its side and supported in saddles the tank could be made of No. 14 gage copper, with joints brazed and copper-riveted.

Engineers' Study Course

Simple Lessons and Lectures for Home Study

PERCENTAGE

When we are told that the price of certain articles is so much "per dozen." "per ton" or "per pound" we readily understand what is meant. The word "cent" is an abbreviation of the Latin *centum*, which means one hundred. Thus we have the century, one-hundred years; cent, the one-hundredth part of a dollar; centimeter, the one-hundredth part of a meter, etc.

Per cent., therefore, means by the hundred or hundredths; a certain number of parts out of each one-hundred parts. Five per cent. of a dollar equals 5 cents; $\frac{5}{100}$ of a dollar = 5 cents.

The following table shows clearly how percentage values may be expressed:

1 per cent. = $\frac{1}{100}$ or 0.01
5 per cent. = $\frac{5}{100}$ or $\frac{1}{20}$ or 0.05
10 per cent. = $\frac{10}{100}$ or $\frac{1}{10}$ or 0.10
20 per cent. = $\frac{20}{100}$ or $\frac{1}{5}$ or 0.20
50 per cent. = $\frac{50}{100}$ or $\frac{1}{2}$ or 0.50
75 per cent. = $\frac{75}{100}$ or $\frac{3}{4}$ or 0.75
100 per cent. = $\frac{100}{100}$ or 1 or 1.
125 per cent. = $\frac{125}{100}$ or $1\frac{1}{4}$ or 1.25
$\frac{1}{2}$ per cent. = $\frac{1}{2}$ of $\frac{1}{100}$ or $\frac{1}{200}$ or 0.005
$\frac{3}{4}$ per cent. = $\frac{3}{4}$ of $\frac{1}{100}$ or $\frac{3}{400}$ or 0.0075

In examples in percentage we have three quantities to consider:

The Base.

The Percentage.

The Rate.

The *base* is the quantity upon which the percentage is calculated, or to which the rate per cent. refers. In the statement "18 is 6 per cent. of 300," the base is 300, because it is the number to which the rate 6 per cent. refers.

When you say 6, 8 or 10 or any per cent. of any quantity or number is so much, that quantity or number spoken of is the *base* because it is the number that is supposed to be divided into 100 equal parts.

Suppose the coal consumption in a plant increased 15 per cent. and that the consumption originally was 100 tons. Then, there is now being consumed 15 tons for every 100 tons or 115 tons. If the consumption had increased 100 per cent., then the amount consumed after the increase would be 200 tons.

The term per cent. is expressed % in writing; 15% is read 15 per cent. In computation or in "figuring" examples in percentage the sign % is not used and the numbers are expressed in decimals instead of in fractions. Thus instead of writing 25% or $\frac{25}{100}$ it is written 0.25.

The *percentage* is that part of the base that is indicated by the rate per cent., and is found by multiplying the base by the rate. Thus in the statement, "18 is 6 per cent. of 300" the *percentage* is 18. Thus in our coal problem: 0.15 × 100 = 15, the percentage is 15.

The *rate per cent.* is the number of 100ths of the base that is taken or considered. As we said above, this figure is always expressed in 100ths in calculations. In the statement "18 is 6 per cent. of 300," 6 is the rate and "6 per cent. of" means 0.06 of the base 300.

If any two of these quantities, that is, the base, percentage or rate are known, the third is easily found. The percentage being the product of the base and rate is easily found by multiplying the base by the rate, while either the rate or base may be found from the percentage by dividing the latter by whatever quantity is known, thus:

$$Base \times Rate = Percentage$$

$$\frac{Percentage}{Base} = rate$$

$$\frac{Percentage}{Rate} = base$$

The chief thing to remember is that the per cent. or rate is to be expressed in 100ths.

To apply what we have learned a few problems will be worked out.

Rule: *Percentage equals base multiplied by rate expressed in 100ths.*

Example: A battery of boilers burned 12 tons of coal per day; after stopping leaks in the settings and changing the manner of firing a saving of 4 per cent. was made. What was the saving in coal per day?

Remember that the 12 tons is supposed to be divided into 100 equal parts.

That a saving of 4 per cent. was made means simply that of every 100 parts of the 12 tons, 4 parts were saved, i.e., the saving is $\frac{4}{100}$ or 0.04 of the amount originally used. Here 12 is the base and 4 is the rate and we want to find the percentage or saving. Then applying the rule:

$$12\ tons \times 0.04 = 0.48$$

or nearly ½ ton saved per day. Ans.

When the percentage and base are given, to find the rate:

Rule: *Divide the percentage by the base; the quotient will be the rate expressed in hundredths.*

Example: A battery of boilers required 12 tons of coal to evaporate a given amount of water. After the scale and soot were removed, 0.48 ton less would evaporate the same amount of water. What was the saving in per cent.?

Here 12 tons is the base and 0.48 ton is the percentage. Now to find the rate.

$$\frac{0.48}{12} = 0.04$$

or 4 per cent. is coal saved. Ans.

When the percentage and rate are given, to find the base.

Rule: *Divide the percentage by the rate, expressed in hundredths; the quotient will be the base.*

Example: Changes in operating a battery of boilers resulted in a saving of 0.48 ton of coal per day, which was a saving of 4 per cent. What was the coal consumption?

If 0.48 ton is 4 per cent., 100 per cent. will be

$$\frac{0.48\ ton}{4} = 12\ tons. \quad Ans.$$

GAIN OR LOSS PER CENT.

When it is desired to find how much the base has been increased or diminished by the addition or subtraction of a certain percentage we may calculate it in either of two ways.

Example: A certain grade of coal contains 11 per cent. of ash. How many pounds of combustible are there in one long ton or 2240 lb. of such coal?

First, we may calculate the percentage of the ash by the method already given, and by deducting the amount of ash from the total or 2240 lb. find the amount of combustible, thus:

$$0.11 \times 2240 = 246.4\ lb.\ of\ ash.$$

$$2240\ lb. - 246.4\ lb. = 1993.6\ lb.$$

of combustible. Ans.

Second, we may say that since the 2240 lb. equals 100 per cent. and 11 per cent. is ash, then

$$100 - 11 = 89\ per\ cent.$$

of the coal is combustible.

Now

$$2240\ lb. \times 0.89 = 1993.6\ lb.$$

of combustible. Ans.

Example: A boiler was "doing" 350 hp. and later another engine was installed which increased the load on the boiler by 15 per cent. How much power did the boiler develop after the increased load?

We may find the additional load by finding 15 per cent. or

$$0.15 \times 350 = 52.5\ hp.$$

and then add this 52.5 to the original load or 350 hp. and so find the total

$$350 + 52.5 = 402.5\ hp.$$

is the total load. Ans.

Or, we may say that as 350 hp. is 100 per cent., the total load after the engine was added was 115 per cent. or

$$1.15 \times 350 = 402.5\ hp.$$

Perhaps the most frequent error in calculations involving percentage is that due to failure to distinguish between the

sum or difference of the percentage and the base.

Suppose you are buying coal that contains 11 per cent. of ash and you wanted to buy enough so that you would have just a ton, 2240 lb. of combustible. How many pounds of coal would you buy?

At first you are apt to take 11 per cent. of 2240 lb. as the additional number of pounds required.

$$1.11 \times 2240 = 2486.4 \text{ } lb.$$

But the ash is not 11 per cent. of the combustible required, but of the unknown quantity of coal which is the *base* from which the percentage is to be computed. The 2240 lb. is the *difference* between the base and the percentage, i.e., between the total amount of coal and the ash that it contains.

The whole quantity or 100 per cent. is made up of ash and combustible, and if the ash is 11 per cent., then the combustible must be 100 — 11 = 89 per cent. Then the required amount of combustible must be

$$\frac{2240 \text{ } lb.}{0\ 89} = 2517 \text{ } lb. \quad Ans.$$

Price and Discounts

The prices of power-plant machines, material and supplies as given in catalogs or price lists are nearly always subject to discounts, that is, the listed price is subject to reduction; a part of it is not to be paid or counted. Usually one cannot correctly estimate costs from a price list without knowing what discounts are allowed as the listed price gives no accurate indication of the actual cost.

Discounts are given in per cent. and expressed, 40 per cent. and 10 per cent. off or "40 and 10" as is usually said. One is apt to think this means 50 per cent. off, but it does not. It means that a discount of 40 per cent. is first made and a discount of 10 per cent. is made on the remainder. If the list price is $2.50 with "40 and 10" off, then 40 per cent. of $2.50 = $1. Now we deduct $1 from $2.50 = $1.50. Then 10 per cent. of $1.50 = $0.15, or 15 cents, and $1.50 — 0.15 = $1.35 = the actual cost.

This same procedure holds true no matter how the discounts are given, i.e., whether it is "40 and 10" or "40, 10 and 5."

Examples for Practice

(1) A pound of a certain coal contains 14,500 B.t.u., but this is 15 per cent. more than is actually used to generate steam in the boiler. How many B.t.u's. then, are used in generating steam?

(2) A plant used 4 tons of coal in one day and 9 lb. of water was evaporated per pound of coal. The feed-water consumption on this day was 25 per cent. greater than on the preceding day. What was the consumption on the preceding day?

(3) An engine was bought for $8000 and later sold for $6000, what was the loss in per cent.?

(4) If 26 tons of coal is 18 per cent. more than was received last week, how many tons were received last week?

(5) The listed price of a 16-in. siphon condenser, circulating pump and engine is $1225, but a discount of "20 and 10" is allowed. What is the actual price?

(6) The condenser auxiliaries require 2 per cent. of the total steam used by the main engine; the engine uses 3600 lb. per hour. How much do the auxiliaries require?

(7) An engine runs at 230 r.p.m. at no load and its governor is supposed to regulate the speed within 2 per cent. from no load to full load. The engine makes 228 r.p.m. at full load; does the governor regulate within 3 per cent.? If not, how "close" is the regulation?

Answers to Last Week's Questions

(1) 641,686 oz.

(2) 66,094 in.

(3) 20 in.

(4) 2.54 cm.

(5) 30.49 × 25 = 762.24 cm.

(6) (a) 6 miles; (b) 9.653 km.

(7) 23 land miles.

(8) 123 circ. in.

(9) 115.14 sq.m. = area.

(10) (a) 27.320 liters; (b) 1,848,000 cu.in.

(11) 1728 cu.in.

(12) 366.2 long tons.

(13) (a) 185 ft. (b) 13.6 ft.

Correction: The answer to No. 3 of the "Example for Practice" in the Engineers' Study Course III, as given in the Nov. 26 issue, should have been 4750 $\frac{76}{261}$ lb. instead of 2790 $\frac{28}{1805}$ lb.

[In order to make the Engineers' Study Course still more interesting, John Ketcham, of Diamond, Ohio, suggests that those following the course communicate with others who are following it. In this way papers could be exchanged and questions asked and answered by those who club together.

Mr. Ketcham thinks that an exchange of papers and ideas in this way would make the course very helpful to those who find it difficult to master the subject of mathematics.

POWER will be pleased to print the names and addresses of those who would like to take up Mr. Ketcham's suggestion. Those who desire to do this should send in their names and addresses and from the printed list select the names of the persons with whom they wish to communicate.—EDITOR.]

Hardened Copper

Our best knowledge of early man and the materials and facilities with which he labored indicate that before he learned to make, harden and sharpen steel, he was mainly dependent upon copper for cutting tools. We think of copper as one of the softest of the metals and one of the last that we should expect to take a cutting edge, and much speculation has been indulged in upon the "lost art" of making it do so.

As a matter of fact, there is no known nor conceivable way to make pure copper other than as it is, nor to doctor it so that its natural ductility will not assert itself when it is subjected to pressure or impact. The cutting tools of the ancients, which apparently did take very fine edges, or which were adequate for such heavy work as stone cutting, etc., were made not of copper but of an alloy of copper, tin and perhaps other elements. The tip of a copper soldering iron will sometimes become so hard that a file will not cut it, through its absorption of other metals with which it comes into contact in its heated condition.

The writer hereof has been using for some time a pair of desk shears, presented to him by S. R. Dawson, of 72 Broadway, New York, and made out of what Mr. Dawson calls "hardened copper," but which he acknowledges is a copper bronze consisting of copper, tin and other ingredients. Their color is that of yellow brass and they hold both it and their edge nicely. Mr. Dawson manufactures a long line of cutting tools, including razors and surgical instruments from this bronze and they take and apparently hold smooth, fine edges.

It is, however, more with reference to its use in engineering applications than in the cutlery trade that our readers will be interested in hardened copper and these applications promise to be numerous. It is claimed that it makes an excellent bearing material, that connecting-rod brasses made of it may be run directly upon the pins without danger of cutting or freezing and several of the large railroads and manufacturing companies are using it successfully in places where metals must move in contact at considerable speeds or under considerable pressure. The inventor would be pleased to communicate with any designer or manufacturer who is involved in a difficult bearing proposition, and he has been able to satisfactorily solve the problem in a number of cases.

Oklahoma City's new $500,000 federal building, which was recently opened, is electrically equipped throughout. The lighting load amounts to 45 kw. and there are about 50 hp. in various motors used for driving, heating and ventilating systems, vacuum cleaners, pump, air compressors, elevators, etc.

Mercury Column Measuring Pressure

By A. O. Doane

Where great accuracy is needed mercury columns are useful. This article describes the operation of U tube and cistern types and their application in measuring steam and gas pressures.

Before the spring type of pressure gage was introduced the mercury column was generally used to indicate pressures. The convenience and cheapness of the spring gage has resulted in the almost complete disappearance of the mercury instrument, except in laboratories and for scientific work, and for use as a vacuum gage.

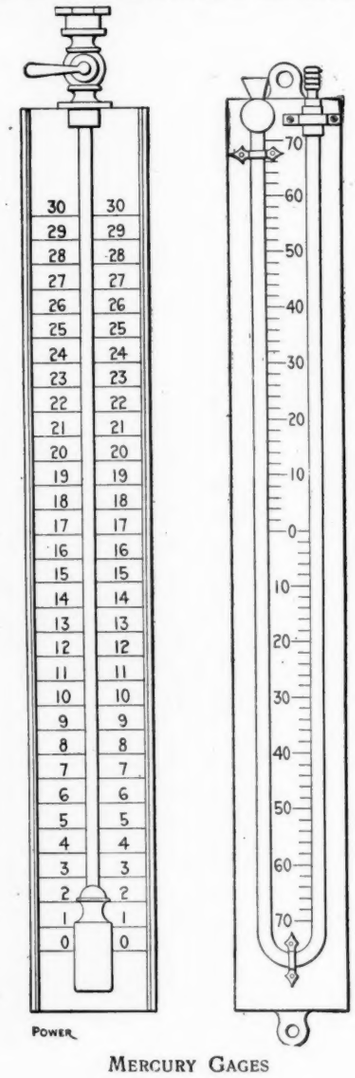
Where great accuracy is necessary, such as in engine, waterwheel and pumping-engine trials, the mercury column has a useful field. It is very sensitive and having no mechanical devices to get out of adjustment, its accuracy is not subject to the variation which is an undesirable feature of the spring gage when used for fine work.

The two principal forms of the mercury instrument are the double-column or U-tube and the single-tube or cistern type. The former is the simplest, consisting of a glass tube bent in the form of a U, or as sometimes made, two glass tubes connected by a hollow iron base piece. The double-tube form is largely used as a vacuum gage in connection with condensing engines.

Mercury is poured into the instrument until it rises in the tubes to the proper height for the range of pressure to be indicated. The scale is commonly graduated with the zero at the middle and in inches above and below this point. The scale is adjustable and the zero mark is set opposite the tops of the mercury columns in the tubes, which are at the same level when there is no pressure on the gage. In operation, one column goes down and the other up, and the difference in level between the tops of the columns is read from the scale in inches of mercury.

The cistern type of instrument consists of a closed reservoir of mercury from which a single tube rises, the bottom of the tube being very close to the bottom of the reservoir. The scale is divided so that the zero shall be the surface of the mercury in the reservoir. When pressure is applied the mercury rises in the tube, and the length of the column above the surface of the mercury indicates the pressure. As the mercury is forced up in the gage tube, the level of the mercury surface in the reservoir falls, and this causes a variation in the elevation of the zero point on the scale, depending upon the amount of the pressure and the ratio of the areas of the reservoir and the gage tube at the point where the upper surface of the column is observed.

There are several ways of dealing with the difficulty caused by this variation in the position of the zero point. One is to compute the interval of the scale, to allow for this variation; another to provide some means, such as glass windows in the walls of the reservoir or a gage-glass fixture attached to the pot in such a way that the level of the mercury surface may be observed. The scale then has an adjusting device, so that the zero mark may be placed level with the mercury surface. If the pressure varies considerably and frequently, this way is undesirable and the first is preferable.



MERCURY GAGES

Still another way is to use a fixed zero point at the level of the mercury surface when there is no pressure on the gage. The scale is graduated in both directions from this point and the readings showing the elevations of the upper and lower mercury surfaces are added to obtain the total length of the column. A gage-glass or other device for observing the elevation of the mercury in the reservoir must be provided.

The length of the graduations of the scale, taking into consideration the drop of the mercury surface in the reservoir as the mercury rises in the gage tube, may be computed by using the following formula:

$$X = \left(\frac{1}{S} \times \frac{1}{\left(1 + \frac{a}{A}\right) - \left(\frac{a}{A} \times \frac{1}{S}\right)}\right) \times 12$$

in which

X = Length of the interval on the scale in inches corresponding to 1 ft. of water;
S = Specific gravity of mercury;
a = Internal area of the glass-gage tube in square inches;
A = Internal area in square inches of the mercury reservoir minus the transverse area of metal in the iron pipe projecting into the mercury in the reservoir and connecting with the glass-gage tube.

The specific gravity of mercury is taken as the ratio of the weight of a unit volume of mercury at the temperature of the column to the weight of an equal volume of water at the temperature of the water, which is the source of the pressure the gage is to measure. As the temperature of both water and mercury is subject to variation, it is well to assume temperatures which will represent as nearly as possible average conditions, and, if greater refinement is desired, to correct the readings for the actual temperatures prevailing at the time the observations are taken.

If the length of scale interval for 1 lb. pressure is desired, the length for 1 ft. of water is multiplied by the quotient obtained by dividing 144 by the weight in pounds of a cubic foot of water at the temperature used in obtaining the specific gravity of the mercury.

The best authorities give the specific gravity of mercury at 32 deg. F. compared with water at the point of maximum density 39.2 deg. F. as 13.5956, and the coefficient of cubical expansion per degree Fahrenheit as 0.00010055. From this the specific gravity at any other temperature may be computed.

For ordinary ranges of temperature the specific gravity will vary from 13.56 to 13.59. Absolute accuracy in practical work is difficult to obtain, as the mercury may contain impurities or the density of

the water may vary, owing to dissolved minerals or absorbed air or other gases.

If it is desired to measure steam or gas pressure, the specific gravity is taken at the temperature of the column of mercury, and the scale may be graduated in pounds per square inch. For steam, due allowance must be made for the effect of the column of condensed water which is generally present in the steam-pressure pipe. The pressure per square inch, due to the height of this column, must be subtracted from the total pressure balanced by the column of mercury, to determine the true steam pressure.

If high pressures are to be measured, the height of the mercury column may prove troublesome as the surface of the mercury in the gage-glass should be at a convenient height above the floor for observation. This may sometimes be provided for by putting the reservoir under the floor, or in a pit or pipe sunk in the ground at the proper elevation and connected with the gage-glass with ordinary ⅛-in. iron pipe. Only sufficient glass tubing need be used to cover the range of pressure expected, with some additional height to provide for surges in pressure. If connected to water-works piping, subject to water rams or large variations in pressure, an overflow pipe should be provided from the top of the glass tube, with a receptacle at the bottom of sufficient capacity to hold the entire amount of mercury in the reservoir. No metal except iron, pure aluminum, solid nickel or platinum should be used where mercury can come in contact with it.

The glass tube may be secured to an iron socket, made to screw on the top of the iron pipe, with a cement made by mixing litharge and glycerin to a paste. This should be allowed to set about 24 hours before putting the gage in service.

If the mercury reservoir is set below the surface of the floor, or in any other location difficult of access, or if no window or gage-glass is provided, the elevation of the surface of the mercury may be determined in the following way:

Take a piece of ⅛-in. iron-pipe-size brass pipe, or plain iron pipe may be used for temporary work, long enough to reach from the surface of the mercury in the reservoir to a convenient point for observation. A rubber-covered copper or aluminum wire is then passed through the pipe, allowing about ¼ in. of bare wire to project from the bottom of the pipe. The wire is securely sealed into the lower end of the pipe, so that no water can enter, and must be well insulated from the pipe. A mark is made near the upper end of the pipe which is at a known distance from the point of the wire. A stuffing-box is provided on the cover of the mercury reservoir, and the pipe containing the wire works through this and may be raised or lowered until the point of the bare wire is in contact with the surface of the mercury. To indicate this contact one or two cells of dry battery are connected in a circuit containing an instrument for indicating a flow of current, such as a voltmeter, battery tester or an electric bell. The wiring is arranged so that the circuit is closed when the point of the wire touches the mercury surface. If a copper wire is used the mercury will cling to the point and the surface at the point be raised a short distance when the wire is raised. With a little practice it is possible to determine the elevation of the mercury surface to within 0.01 in. Knowing the distance from the mark to the point of the wire, the elevation of the mercury may be easily found.

This operation may be carried out under water pressure sufficient to raise the mercury to approximately the normal level in the glass tube, in which the top of the mercury column is observed. The distance from the lower mercury surface to the mark on the pipe plus the distance from this point to the upper mercury surface in the glass tube will give the length of the mercury column balancing the pressure.

By making a mark opposite the top of the mercury column and computing what the measured length of column represents in the units used for the graduations, the scale may be conveniently and accurately adjusted.

If the pressure varies considerably while these observations are being made, the pressure valve should be shut off, leaving the mercury in the reservoir under pressure. All joints and the stuffing-box must be tight if this is done; otherwise the mercury in the tube will rapidly drop.

A pressure pipe from a water main, or discharge pipe of a pumping engine should never be connected to the top of the pipe but always to the side or lower part, to avoid the entrance of air. All summits on the pressure pipe leading to the mercury reservoir should have air cocks, which must be opened from time to time to see if air is collecting.

With considerable fluctuation in pressure, using a small pressure pipe, ¼ or ⅜ in. is advisable. Throttling a valve or cock closely, to damp the movement of the column should not be attempted, but a large diameter glass observation tube should be used; one ¾-in. internal diameter gives good results in most cases. If any throttling is necessary, a cock gives better results than a valve.

In cleaning the glass tube, no metal should be allowed to touch the interior surface, as it will cause the tube to crack. A few drops of white-wine vinegar on top of the mercury in the tube will greatly assist in keeping the glass clean and bright.

If the mercury gage is correctly installed and given the small amount of care it requires, it will be found very accurate and satisfactory for work to which it is especially adapted.

Causes of Notable Boiler Explosions

Probably the most disastrous explosion which ever occurred was that of a horizontal tubular boiler in a large shoe factory at Brockton, Mass., on the morning of Mar. 20, 1905. One part of the building became a mass of ruins, which took fire, and 60 of the 400 persons employed in the factory were either immediately killed or buried alive, and over 100 were more or less seriously injured.

Investigation revealed that the boiler burst as a result of a defect known as a lap-joint fracture. This is one which extends along the longitudinal seams of lap-jointed boilers, and is due to imperfect rolling of the plates when curving them. Sometimes the last few inches at each end slipped off the rolls and did not become bent to the true form. Then they had to be forced to the desired radius, which was often done with sledge hammers, the plate being placed over a "set."

Such obviously severe treatment seriously stressed the plate, and as, in addition, the true form was not obtained even after treatment, the subsequent riveting of the ends together tended to set up fracture. Some ignorant or reckless boiler maker would drill or even punch the holes prior to bending, and, under such circumstances, it is no wonder fracture developed. Unfortunately, the lap-joint fracture usually occurs on the inside of the overlapping plates, where it cannot be detected, and hence may develop while working until the boiler is weakened to the bursting point.

The boiler which failed at Brockton was periodically examined by an insurance company, and at the last inspection no defects were found, so that the boiler was apparently safe for the stipulated working pressure. This explosion did much toward doing away with the lap joint, and in the present best practice only butt joints are used for the longitudinal seams. With such joints the true cylindrical form of the shell is retained and any tendency to straining, either in constructing or working, is avoided. Where lap joints are still used, rolls are employed which are capable of rolling the plates clear to their ends.

Another disastrous explosion was that which occurred on board

H. M. S. "THUNDERER"

in 1876, when 45 were killed and 33 seriously injured. This accident was due to the safety valves becoming inoperative, through faults in workmanship, such that when the wings expanded with the heat they became jammed. Probably this explosion would have been averted,

if the boiler pressure gage had had a stop pin at the high-pressure end of the dial. At the time important steam trials were being conducted, and in the excitement and bustle, it escaped notice that the pressure-gage pointer went completely around the gage and on past the zero mark again, so that it registered an apparently low pressure, when the pressure had reached the bursting point.

An account of a terrible boiler explosion appeared in the *Vulcan* (the monthly organ of the Vulcan Boiler & General Insurance Co., of Manchester, England) for August, 1905. It stated that one of the boilers on board the

U. S. Gun-boat "Bennington"

which was lying in the harbor of San Diego, Calif., burst and was immediately followed by the explosion of the powder magazine. All on board were either killed outright or horribly maimed, the dead and missing amounting to 60 and the injured to 75. According to this report, one of the port boilers had become defective and been recently repaired, and was supposed to have failed first, causing the others to fail and igniting the powder magazine.

Shamokin Explosion

Many instances are on record of whole ranges of boilers exploding, but in no case were the results so disastrous as those above related. On Oct. 11, 1894, at Shamokin, Penn., 27 boilers out of a range of 36 burst, but fortunately only five persons were killed and a few injured. These were very long plain cylindrical boilers externally fired. This type is especially likely to fail from a defect commonly known as a seam rip. Being externally fired, the bottom of the shell is directly exposed to the intense heat of the furnace gases, while the top remains comparatively cool. The bottom consequently expands much more than the top, excessively straining the circumferential seams, which gives rise to fractures or seam rips extending for various distances round the seams. Fortunately, seam rips usually occur when the boilers are being cooled down for cleaning or inspecting, but occasionally they occur without warning when the boilers are at work, and not uncommonly the fracture extends completely around the shell; then unless the boilers are well stayed longitudinally, a violent explosion results.

Explosion at Redcar, England

On June 14, 1895, a similar explosion occurred at Redcar, in Yorkshire, England, when 11 out of 15 boilers exploded, killing 12 persons, and injuring nine others. These boilers were nearly 70 ft. long ,and the company which insured them had reported quite a number of seam rips and advised cutting the boilers into two parts and lashing the ends together by longitudinal stays to minimize the risk of failure. Unfortunately, the advice was not acted upon. The danger of these long externally fired boilers has been long recognized and happily this type is now fast becoming obsolete.

Another Multiple Boiler Explosion

occurred at Friedenhutte, Germany, in July, 1887, when a whole range of 22 boilers gave way, killing 12 people and injuring 35 more. These boilers were of the ordinary elephant type, fired by blast-furnace gas. It is supposed that an explosive mixture of blast-furnace gas and air collected in the flues and became ignited and that the shock from this explosion caused the boiler to let go.

A Water-tube Boiler Explosion

Water-tube boilers are commonly supposed to be immune from serious explosion, but this is fallacious. In December, 1907, at St. Louis, a whole range of seven such boilers exploded, six of them being blown to pieces, while the remaining one had a large number of tubes blown out. Eight people were killed and 21 seriously injured. According to *Locomotive,* one of the tubes of the first boiler burst, but whether as a result of scale, overheating, or other defect it is not known. This tube bursting probably tore out the others, until the whole boiler was blown to pieces, which doubtless led to the explosion of the next boiler, and so on.

At present boiler explosions are less frequent than before the advent of the various boiler-insurance and inspection agencies, and the more vigilant inspecting of boilers, both during construction and working. Really disastrous explosions are comparatively rare, and such as do occur, are usually from carelessness and neglect, or even recklessness, as was the case with the explosion at York, Penn., Aug. 10, 1908. This explosion, a full account of which was given in Power, Aug. 25, 1908, was due to sheer recklessness. The boiler was an old one, the plates were badly wasted and cracked, neither pressure nor water gage was attached, and the safety valve was practically useless, having been known to blow off but once in 18 years. A boiler worked under such conditions should be expected to explode. Steam boilers now are rarely worked without suitable fittings, and generally are periodically examined by experts, so that defects are discovered and corrected before they become serious.

Every year, knowledge of both the construction and working of steam boilers becomes more and more thorough, and with the exercise of care and skill on the part of those operating the plant, together with periodical and vigilant inspection by experts, boiler explosion should soon become almost a thing of the past.

A little book entitled "The Distington Multiple Boiler Explosion and Other Multiple Boiler Explosions," recently published by the chief engineer of the National Boiler & General Insurance Co., Manchester, England, gives a full account of an investigation into the cause of the explosion of four boilers at a large iron works, together with an account of 17 other multiple explosions. Those interested in the subject will find it well worth reading.

Worcester Associations Celebrate N. A. S. E.'s Thirtieth Anniversary

On Saturday, Oct. 26, the members of Worcester Association No. 4 and H. E. Stone Association No. 32, both of Worcester, met in the State Mutual Building banquet hall and with their invited guests celebrated the thirtieth anniversary of the National Association of Stationary Engineers and the tenth anniversary of H. E. Stone Association No. 32. Under the direction of a joint committee, Messrs. Tolman, Lambert and Stockwell, Worcester No. 4, and Messrs. Wilder, Seed and Miller, No. 32, a delightful banquet was provided, with music.

About 100 members and guests attended, among those present being Mayor O'Connell, of Worcester; State President Maloney, Fall River; Past President Wheeler, New York; Past State President H. E. Kelsey, Lowell; State Inspector of Boilers James B. De Shayo and State Inspector Morton, of the Worcester district; H. E. Stone, consulting engineer, and Ole Petersen, state secretary.

After the repast, Toastmaster Seed introduced Mayor O'Connell, who proposed the toast, "The N. A. S. E." and spoke briefly. Speeches were also made by Past National President Kelsey, State Secretary Ole Petersen, State Inspector De Shapo; State President Maloney, Mr. Hayes, of the State Educational Committee, and C. J. Wilder.

The meeting was one of the most enjoyable in the history of the N. A. S. E. in Massachusetts.

Brush Bar Belt Dressing

"Brush Bar" belt dressing, made by the Wayo Manufacturing Co., Buffalo, N. Y., is put up in cylindrical form, one full pound of dressing to each bar. The bar is 2 in. in diameter. Each stick of dressing is made with a 2-in. leather brush end for removing dirt from a belt while running. Then the stick is reversed and the dressing is evenly applied to the belt. The brush serves as a handle to the bar of dressing, thus permitting of using the entire stick without waste.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

The Aktiebolaget Scania-Vabis, of Stockholm, Sweden, has decided to enlarge its plant. Good! We congratulate the—er—them, or it. From our ignorant point of view, to enlarge the name would be a mistake.

Ezra Semple's Silas, who is firing in a "juice factory" down to New York, writes Ezry that he's tickled because the courts have swatted the "bathtub trust." Si asseverates that New York is so ripped up and dusty that a bath, even in cold weather, is an absolute necessity. The city folks haven't anything on our Si.—*Whangville Banner.*

A newspaper calls this " a little story full of interest," though it makes a noise like a "movy": A Chicago engineer, out of work, hungry and discouraged (orchestra plays shivery music), saw a hurrying citizen drop a real cent and "scorn to pick it up"—just like that. Did the engineer pick up the cent and buy a turkey with it? NO! He bought a paper, "scanned the want ads, picked a job as a hotel engineer and went to work without saying anything." This paper should have spelled that "cent" with an "s" before the "c." Is it for this that we have engineers?

"Our leading townsman Zebulon Tucker has no equal in Four Corners when it comes to practicing economy," says the *Trumpet*. "He took his electric fan apart last week for cleaning purposes. When he got it together again he had 23 pieces left over."

Every now and again some trouble-hunter tries to upset business by prophesying hard times for us working people. Old man Trouble won't trouble us much if we take the trouble to cock one eye up at old Sol and keep a-inchin'. Ding bust a pessimist anyhow!

(The bean-oil industry in South Manchuria has greatly developed in recent years.—Consular Report.)

Don't you remember those sweet Alice days, Ben Bolt, when we oiled our "bean" before hieing away to Alice? How each spear was oilingly cajoled to make a showing? How redolent was the odor we disseminated as we entered the vine-clad cottage? Them was the happy days, Ben! Today, alas, we regretfully gaze on that ingrowing forehead and sigh: "Oh, where are the spears of yestere'en?" And oil did it, though we can't prove it.

NEW PUBLICATIONS

STEAM TURBINE DESIGN. By John Morrow. Published by Longmans, Green & Co., New York. Cloth; 471 pages, 5¾x9 in.; 150 illustrations; plates; tables. Price, $4.50.

This book deals chiefly with the design of reaction turbines with especial reference to marine application by British ship builders. There are, however, numerous articles on turbines for electrical and other land purposes.

The author first takes up the fundamental principles of design, showing the application of entropy and other heat diagrams to the subject; this is done in an exceptionally clear manner and with the absence of complicated mathematics. This is followed by chapters on design of the various parts which make up the machine; formulas are derived for calculating these parts together with numerous examples of existing installations in which the applications of the formulas are illustrated.

The Rateau and other types of regenerators are described and their application to low-pressure turbines discussed.

Under the design of propellers for turbine-driven ships methods are given for proportioning propeller and turbine speeds so that a maximum efficiency for the unit will be obtained.

Several methods of speed reduction are described, including: Melville-McAlpin gears, hydraulic and electrical transmission systems and their points of advantage are discussed.

Chapters are included on the general principles of surface condensation, condenser design, and the work is concluded by a brief description of the impulse turbine.

Anyone interested in turbines will find this book is well worth reading. It is especially valuable to students and designers of marine turbines.

LIGHT, ITS USE AND MISUSE. A primer of illumination prepared by the Illuminating Engineering Society, 29 West Thirty-ninth St., New York. Paper; 20 pages, illustrated.

This little pamphlet has been prepared primarily for the users of artificial light. With this purpose in view technicalities have been avoided and the text confined to a simple exposition of the general principles of illumination. With the aid of a number of well chosen photographs the proper and improper distribution of light is shown for different kinds of service, both in the home and in the factory, and special attention has been called to the evils attending bad lighting.

BOOKS RECEIVED

EXPERIMENTAL ENGINEERING. By R. C. Carpenter and Herman Diederichs. John Wiley & Sons, New York. Seventh edition, rewritten and enlarged. Cloth; 1132 pages, 6x9¼ in.; 636 illustrations; indexed. Price, $6.

COAL. By E. E. Somermeier. McGraw-Hill Book Co., New York. Cloth; 175 pages, 6x9½ in.; illustrated; tables. Price, $2.

PRACTICAL MATHEMATICS. Part I and Part II. By Claude Irwin Palmer. McGraw-Hill Book Co., New York. Cloth; 291 pages in the two volumes; each 4½x7 in.; tables; 150 illustrations. Price, 75c. each.

HYGIENE FOR THE WORKER. By William H. Tolman. American Book Co., 231 pages, 5x7½ in.; illustrated.

THE "MECHANICAL WORLD" POCKET DIARY AND YEAR BOOK FOR 1913. Published by "The Mechanical World," Manchester, England. Cloth; 439 pages, 4x6½ in.; illustrated. Price, sixpence.

TOOTHED GEARING. By George T. White. D. Van Nostrand Co., New York. Cloth; 226 pages, 4½x7½ in.; 136 illustrations; tables. Price, $1.25.

THE THEORY OF ENGINEERING DRAWING. By A. A. Adler. D. Van Nostrand Co., New York. Cloth; 312 pages, 5½x9 in.; 273 illustrations. Price, $2.

Large Suburban Contract at Minneapolis

The Minneapolis General Electric Co. has closed a 10-year contract for selling electric current covering the requirements of the Northern Power Co., which operates under 25-year franchises in Excelsior, Deephaven, Tonka Bay, Wayzata and a number of other communities on the shores of Lake Minnetonka, near Minneapolis.

To serve this desirable business, including at present a population of about 12,000 in summer and 5000 during winter, a nine-mile continuation of the transmission line now being extended to Hopkins, Minn., will be necessary. The Northern Power Co. is rapidly extending its lines to all of the summer resorts around the lake. It will close down its steam-generating station at Excelsior, 18 miles from Minneapolis, and many small isolated lighting plants will be displaced by electricity, as well as gasoline plants heretofore used for pumping water.

PERSONAL

D. G. Baker, formerly works manager of the Olds Motor Works, at Lansing, Mich., and of the Blake & Knowles Steam Pump Works, at Cambridge, Mass., has been appointed works manager of Carels Fréres, Ghent, Belgium, the largest exclusive builders of the Diesel engines in Europe at the present time. The plant is being doubled. Mr. Baker has just returned from a trip to Ghent, where he concluded arrangements, and will sail in a few days with his family to Europe.

The Engineers' Club, of Boston, is progressing at a rate which meets the expectations of the most sanguine of its organizers. With the election of the candidates, whose names are now before the membership, the resident membership of the club becomes nearly complete. The club house is nearly finished and it is hoped that it will be in readiness for the members' use about the first of January.

Moments with the Ad. Editor

The most necessary quality in a good ad writer is that much-misunderstood quality—*imagination.*

Imagination is quite the opposite of a common conception thereof. It has nothing to do with fancifulness, with "make-believe," with dreaminess.

Imagination means insight. The great scientist who, after gathering data for years, with but a vague idea of the tendency of his investigation, suddenly, with his facts still incomplete, realizes the underlying meaning and unity of it all, as Darwin did in his formulation of the theory of evolution,—*he* has that precious quality, imagination.

The engineer who, studying for months the problem of spanning a great canyon, suddenly sees the finished bridge in his mind's eye—beautiful, complete,with his great problem solved through a new combination of engineering principles—he, too, has imagination. So the statesman who foresees the mental temper of a nation, or the great general who, from a study of circumstances that would baffle, confuse, and make the ordinary leader temporize, sees the one right stroke to win his battle, and strikes.

Imagination in the advertisement writer means insight—vision—appreciation—of the point of view of another. To imagine the "lead" that will get your special audience's attention, or the attention of the greatest number—the point of view that will attract, and the line of argument that will convince, is the one indispensable gift.

The salesman, too, imagines his buyers' view point, and adopts different styles of attack for each case.

The ad writer is compelled to do even more. He must get the *combined* point of view of many—that is, the one element in which thousands of the possible number out of thousands are at one *for his purpose.* He cannot adapt himself tactically to each individual case. He must strike the individual appeal that is most universal at the moment. In other words he, if he be successful, talks to thousands in a way that gives each a feeling that he talks to him personally.

When you see an advertisement in "Power" or some other paper that "gets you" and that is simultaneously "getting" a thousand or ten thousand other men, differing widely from you in a thousand ways, remember, that though it may have been written rapidly and easily by its writer, and may seem a "cinch" for almost anyone to do, yet would probably be the most difficult if not impossible thing for the man without the gift and the training, even if he knew the product in question from A to Z.

Like all simple things into which a special, individual, rare ability goes, it seems easy for the outsider, and is anything but that. There's an incident of a man who had a circular letter to be written. Finding that he was unequal to the task himself, he went to an advertising man, who wrote the letter and charged him $15 for the job

Quite enraged at this impudent and unwarranted piece of highway robbery, he at once appealed to one of the leading advertising authorities of New York, a man who had spent hundreds of thousands of dollars in advertising.

"Say," said he, "how long should it take a man to write a circular letter?"

"Well," replied the big advertiser, "A man of exceptional ability might possibly complete the letter in a day's time, a man of ordinary ability could write it in half a day, and any damn fool could write it in half an hour."

Nine-tenths of the work involved is plain "digging" —digging deep down into the proposition to get at the point of contact between product and purchaser. This is where the gift of imagination—insight—plays its part.

If you find the ads in "Power" interesting and informative, it may be, perhaps, because the men who write them have the gift and the training.